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DOE/NASA CONTRACTOR REPORT

DOE/NASA CR-150849

PASSIVE THERMOSYPHON SOLAR HEATING AND COOLING
MODULE WITH SUPPLEMENTARY HEATING (Quarterly Report)

Prepared from documents furnished by

Sigma Research, Inc.
2950 George Washington Way
Richland, Washington 99352

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For the U. S. Department of Energy

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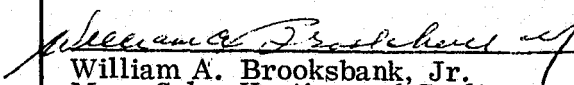


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16. ABSTRACT This report is a collection of three quarterly reports from Sigma Research, Inc., covering progress and status from January through September 1977. Sigma Research is developing and delivering three heat exchangers for use in a solar heating and cooling system for installation into single-family dwellings. Each exchanger consists of one heating and cooling module and one submersed electric water heating element. Some retyping and reformatting were done for clarity.					
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FIRST QUARTERLY REPORT

1.0 INTRODUCTION AND SUMMARY

During the first quarter of development efforts have been directed primarily at determining the required configuration for components to be used in the prototype Maxi-Therm S-101 module. These investigations have centered on three main areas: the exchanger module, the exterior cabinet and the shut-off valve.

With regard to the first item, the exchanger module, extensive analytic studies were carried out to define the best size and liquid circuiting arrangement for the horizontal heat exchanger. This required detailed, digital computer-aided analysis of not only the exchanger proper, but also the liquid transfer system and piping interfaces. The result of this study was the identification of a module possessing both compact size and acceptable performance.

The second area of development, that concerning the exterior cabinet, was carried out in parallel to exchanger development to assure production of a total, functional unit. Some of the considerations in this area of development included consumer appeal, inexpensive production, and compatibility with both scheduled and unscheduled maintenance operations.

In the design of the shut-off valve, which represented the third area of activity, investigations were initially oriented toward selection of valve type; that is, water circulation or air circulation shut-off. From these studies, the air-side valve was found to be superior and further studies were carried out to define the exact configuration. During this latter study, consideration was given to both active (motor operated) systems and passive (differential pressure operated) systems, with the passive unit being chosen.

In Section 2, which follows, the details of studies relating to the exchanger design are contained, with Section 3 going on to cover the exterior cabinet development. Section 4 contains the complete quality assurance plan for this program and Section 5 details the development of the shut-off valve.

2.0 MAXI-THERM EXCHANGER MODULE

The performance of the Maxi-Therm system is determined primarily by the water-to-air exchanger module and, as such, a major portion of the activity this quarter has been centered on refining the design of this part of the system. As discussed with MSFC Technical Monitor J. Hankins previous to, and during, the Preliminary Design Review, the exchanger module has been reoriented to a horizontal, rather than vertical, position. This re-orientation has necessitated both modifications in the method of analytical modeling as well as a change in the number of system variables taken into consideration. In particular, the pressure drops associated with various portions of the system are of much greater importance, with Section 2.2 treating this area in detail. In addition, Section 2.2 also contains the test results and data correlation for liquid side heat transfer, as well as the modeling techniques used for the air side modeling of thermal performance.

Prior to this in-depth consideration of heat exchanger operation, it is beneficial to consider the advantages associated with the re-orientation of the exchanger module. In the following section, 2.1, a number of topics related to this modification are discussed.

2.1 Comparison of Vertical and Horizontal Thermosyphon Operation

In vertical thermosyphons operating in the heating mode (Figure 2.1), water enters at the top, flows vertically downward through the heat transfer elements, and exits at the bottom of the exchanger prior to reentry into the thermal storage tank. During its downward movement, the water gives up heat to the air, thus falling in temperature and rising in density. The pressure which drives the water flow in the system then originates due to the density differential existing between the water in the heat transfer elements of the exchanger and the storage tank, with

$$\Delta P_D = \int_0^h g \cdot \rho(x) dx - gh\rho_T \quad (\text{dynes/cm}^2) \quad (2-1)$$

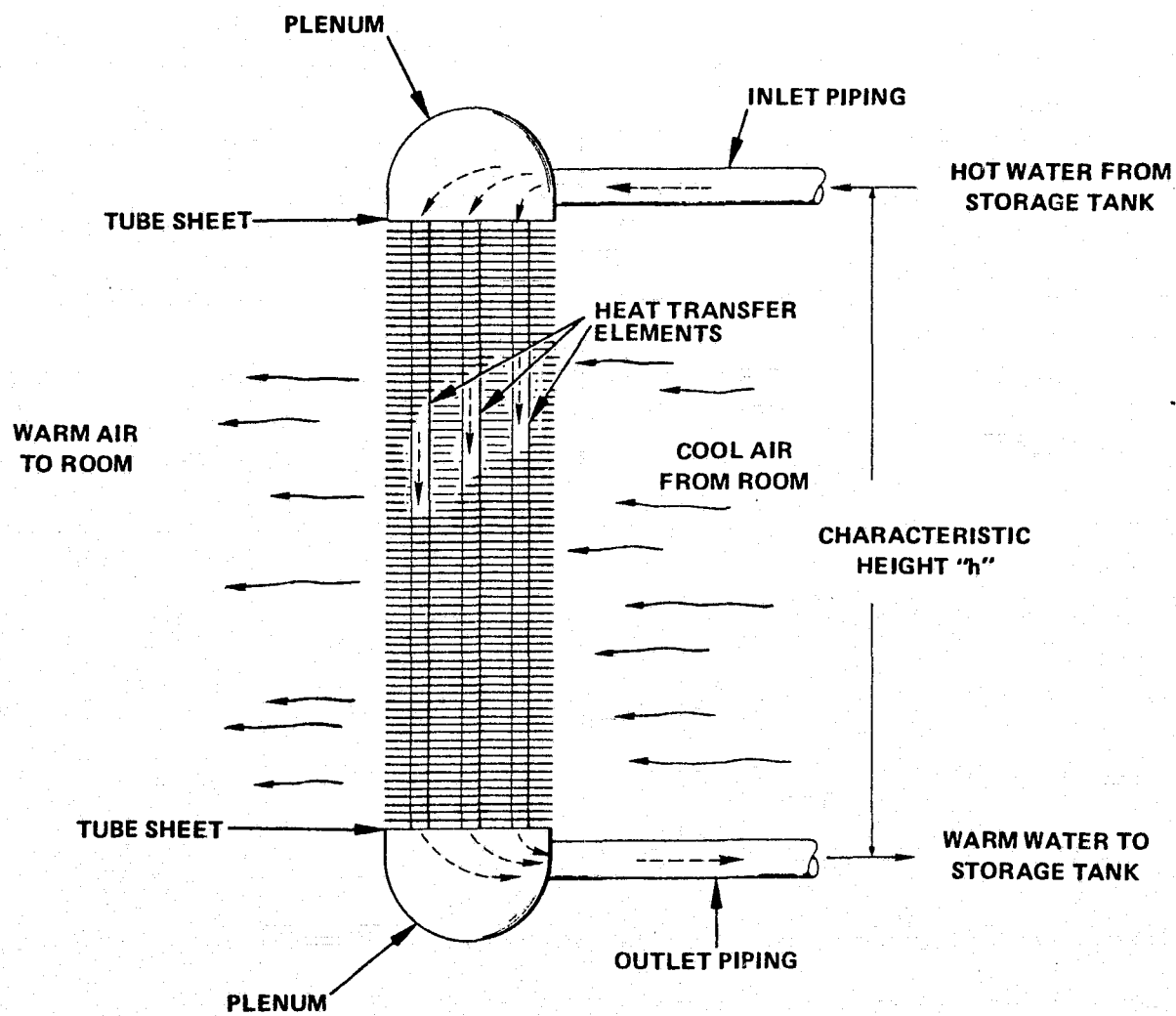


FIGURE 2.1. BASIC DESIGN OF A VERTICAL THERMOSYPHON HEAT EXCHANGER FOR USE WITH LIQUID THERMAL STORAGE TANKS.

where

ΔP_D = driving pressure difference, dynes/cm²

g = gravitational constant, 981 cm/sec²

$\rho(x)$ = position dependent water density in heat transfer elements, gm/cc

h = characteristic system height, cm

ρ_T = density of storage tank water (non-stratified), gm/cc

If the water exiting from the heat exchanger is assigned the density ρ_E , then, in general, the integral $\int_0^h \rho(x) dx$ will be some fraction, α , of the product $\rho_E \cdot h$. This fraction, α , which is always less than 1 in normal systems, can be substituted into Equation 2-1 to yield the expression

$$\Delta P_D = gh(\alpha \rho_E - \rho_T) \quad (\text{dynes/cm}^2) \quad (2-2)$$

In a horizontal thermosyphon (Figure 2.2), a somewhat different driving pressure expression is realized, since the rise in density of the water occurs during flow in a horizontal plane. Before entering the vertically oriented outlet piping, which connects the exchanger outlet to the bottom of the tank, the water has already reached its maximum density. The value of α in this type of system is thus 1, and, if the values of h , ρ_E , and ρ_T are the same, a greater driving pressure differential and, consequently, a higher mass flow rate, are realized.

An additional advantage of the horizontal exchanger position relates to the circuiting of liquid flow in the heat transfer elements. Whereas the vertical unit is constrained to use only multiple parallel paths connected to common plenums at each end, the horizontal unit may utilize totally parallel or parallel-series circuiting. In Figure 2.3, a comparison is shown between the flow in a total parallel system and in one type of parallel-series arrangement. As shown in this figure, all tubes in the parallel flow system connect

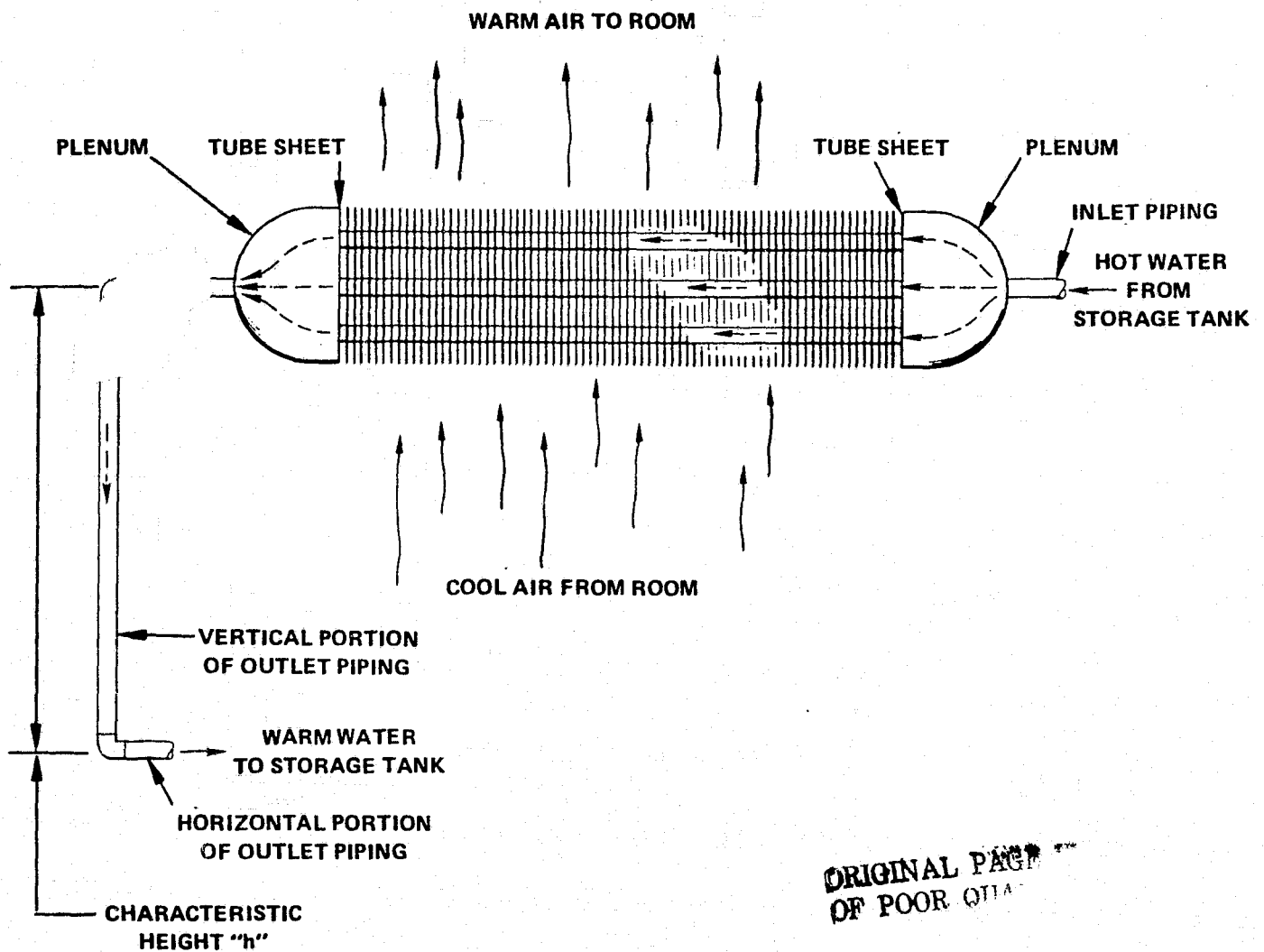
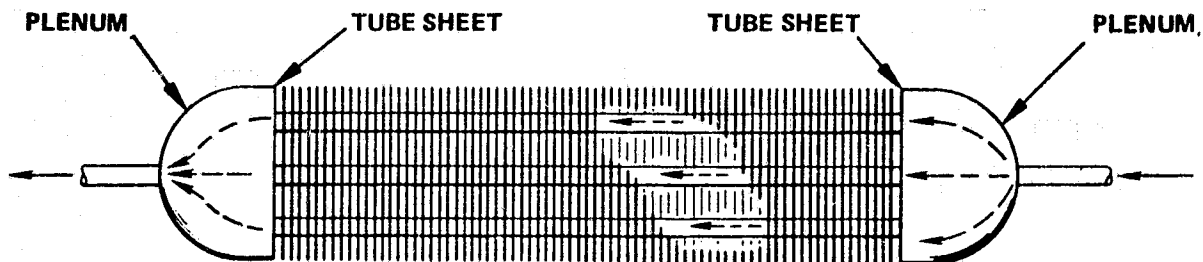
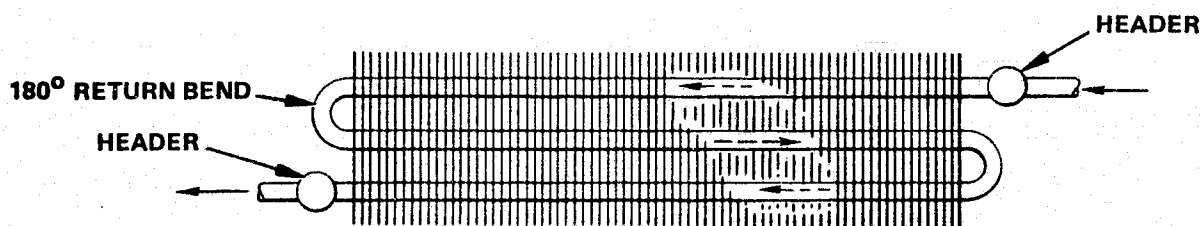


FIGURE 2.2. BASIC DESIGN OF A HORIZONTAL THERMOSYPHON HEAT EXCHANGER FOR USE WITH LIQUID THERMAL STORAGE TANKS.



HORIZONTAL THERMOSYPHON WITH PARALLEL FLOW CIRCUITS



HORIZONTAL THERMOSYPHON WITH PARALLEL-SERIES FLOW CIRCUITS

FIGURE 2.3. TWO POSSIBLE FLOW CIRCUITS FOR HORIZONTAL THERMOSYPHONS. IN BOTH UNITS, ADDITIONAL ROWS OF PARALLEL CONNECTED CIRCUITS EXIST, DIRECTLY BEHIND THOSE SHOWN (INTO THE PLANE OF THE FIGURE).

to a common plenum by means of a tube sheet, whereas the parallel-series unit uses an inlet and outlet header together with 180° bends to facilitate coupling.

If the pressure drop through series connected tube elements is not prohibitive, then this method of circuiting presents a real potential for cost savings in production units. This is predicated on replacing a relatively complex, formed tube sheet and plenum with simple U-bends and a round header, which can connect directly to the outlet piping. In addition, the parallel-series arrangement permits a lower mass flow rate in the transfer system (inlet and outlet piping) when the mass flow rate in the heat transfer elements is of equal magnitude to those in a total parallel system. This, of course, is predicated on an equal number of tubes and vertical tube rows being present in both systems.

Should the parallel-series arrangement be capable of operating at the performance levels required, then the decrease in transfer system mass flow will permit additional improvements, such as allowing the use of longer horizontal and vertical connections to the tank. The former change may be necessary in some installations to permit easy siting of components, whereas the latter can provide both easier connection to the tank as well as increased performance. Effectively, an increase in length in the vertical outlet section increases the value of "h" in Equation 2-2 and provides more driving pressure, which in turn raises the mass flow rate and increases the heat transfer coefficient which exists between the water and tube elements.

In considering the arrangements possible, it is difficult to determine a priority which of the units will be most attractive in terms of cost and performance. Detailed consideration of each part of the system is a necessary first step, followed by computer simulation of the entire system. In the following sections, this has been the approach utilized.

2.2 Exchanger Module Analysis and Experimental Testing

In the design of total forced flow heat exchangers, primary consideration is generally given to the convective coefficients for the two heat transfer media, with secondary consideration being given to the system pressure drops. In the design of liquid side thermosyphon heat exchangers, however, the pressure drop in the liquid system is of comparable importance to the heat transfer coefficient since it ultimately determines the value of this coefficient.

It is necessary, then, to accurately define the relevant pressure drops in all portions of the system and determine their dependence on various system parameters, particularly mass flow rate. Concurrent with this investigation, it is also necessary to determine the effect of mass flow rate, and other parameters, on the liquid side heat transfer coefficient. With the findings from these two investigations, it is possible, using known characteristics of air side performance, to formulate a digital computer code to permit exchanger simulation and optimization.

In the section which follows, 2.2.1, the pressure drop characteristics for the liquid system are detailed, with Section 2.2.2 going on to describe the experiments and data evaluation necessary to define liquid side heat transfer coefficients. Air side performance, from both a thermal and pressure drop standpoint, are covered in Section 2.3.3, while Section 2.2.4 describes the digital computer model and presents results from the parametric studies on exchanger design.

2.2.1 System Pressure Drops

In general, pressure drops arise in a piping system from a number of sources, including expansions, contractions, fittings, valves, corners, and straight-pipe viscous flow losses. In the horizontal thermosyphon there are no fittings or valves, with the shut-off control having been moved to the air stream. Expansions and contractions are present, primarily at the tank entrance and exit and at the plenum-heat transfer element interface.

The plenums and inlet and outlet tubing, being of similar diameter, do not contribute any substantial expansion or contraction pressure losses. Two corners exist in the transfer system, one at the exit from the outlet header and one in the outlet flow piping system, the second being located at the point at which the vertical outlet flow is turned to flow horizontally for return to the storage tank. Both corners will contribute losses to the transfer system, in addition to the expected pressure drops found in straight portions of the inlet and outlet piping. In the fin tube elements, pressure losses also arise from normal viscous flow losses and, for the parallel-series arrangement, in the 180° return bends.

In the following section (2.2.1.1) the trade-offs in inlet and outlet piping are detailed, with sections 2.2.1.2 and 2.2.1.3 going on to discuss the design of the tank outlet and inlet, respectively. The final portion on pressure drop (Section 2.2.1.4) considers the pressure drops associated with the heat transfer elements.

2.2.1.1 Transfer System Design

The inlet and outlet piping represents an area of greater importance for horizontal thermosyphons than for comparably sized vertical units in that the primary heat transfer elements do not form the vertical link between tank exit and inlet. For heating-only units, this causes an increase in the exchanger outlet piping length and correspondingly higher transfer system pressure drops. In addition, elbows are also present in the transfer system for horizontal units to facilitate flow to the storage tank.

To minimize pressure drop in the piping, and thus increase mass flow rate, it is advantageous to both increase pipe diameter and decrease piping length. A decrease in outlet pipe length, in the case of the vertical portion, however, is not desirable since it is this length which directly determines the thermosyphon driving pressure. A decrease in the horizontal portion is likewise undesirable, since sufficient distance must be provided between the

Maxi-Therm unit and the storage tank to permit the installation of pipe couplings. The diameter, then, is the best variable to change if a lower transfer system pressure drop is required. A change in this variable also produces the most effect on piping pressure loss, since pipe pressure drop is given, according to Schlichting⁽¹⁾, by

$$\Delta P_s = 0.24113 \frac{L \cdot \dot{m}^{1.75} \mu^{0.25}}{D^{4.75} \rho} \quad (\text{dynes/cm}^2) \quad (2-3)$$

in the Reynolds number range 5,000-30,000. In the above equation

ΔP_s = straight pipe viscous pressure loss, dynes/cm²

L = pipe length, cm

$\dot{m}$ = mass flow rate, gm/sec

μ = fluid viscosity, poise

D = inside diameter, cm

ρ = fluid density, gm/cc

The Reynolds number range of applicability for equation 2-3 includes operating conditions up to and exceeding the Subsystem Performance Specification rating point for most exchanger configurations considered.

An increase in the internal diameter at the elbow has a similar effect to that of the straight pipe. This relationship, as given in a somewhat different form by Schlichting,⁽¹⁾ is

$$\Delta P_E = 0.3577 \cdot \frac{R^{0.9} \dot{m}^{1.8} \mu^{0.2}}{D^{4.7} \rho} \quad (\text{dynes/cm}^2) \quad (2-4)$$

where

ΔP_E = total elbow pressure drop, dynes/cm²

R = radius of curvature of elbow, cm

and the other terms are as previously defined.

Although equation 2-4 appears to indicate an increase in system pressure drop with increasing elbow radius of curvature, this is not entirely correct. Should this variable be increased, a finite portion of both the horizontal and vertical piping is now contained within the elbow, and thus vertical and horizontal piping lengths are decreased.

Opposing an increase in diameter to permit pressure loss reductions is the material cost factor. In general, the price for components of the diameter considered for the Maxi-Therm units, 7.62 - 10.16 cm (3-4 inches), is dependent upon the weight of the item. If wall thickness is held constant, the increase in price is then directly proportional to the diameter. This rule has been found to be true for both brass and copper components, and approximately correct for CPVC items as well. The latter material, CPVC, is of particular interest since its use would permit reductions in module weight and significant cost reductions ($\approx \$10$ - $\$15$) in piping system costs. Presently, investigations are continuing on its long term stability and strength characteristics when subjected to the stresses and temperatures anticipated in service.

2.2.1.2 Tank Exit Pressure Losses

At the point of exit from the tank, the liquid moves from a relatively large reservoir having small liquid velocities to a tube in which the velocity has a finite value. In doing so, a portion of the static pressure head is converted to velocity pressure head. In addition, when a square-edged tank connection is utilized, a vena contracta, or flow contraction, occurs and total pressure is further decreased. This loss can be significant in that the pressure loss equals one-half of the velocity head ($\frac{1}{2} \cdot \rho V^2 / g_c$).

By utilizing a tank exit connection having rounded corners, this pressure drop can be substantially decreased, in that the flow conforms to the tube wall and the vena contracta is eliminated. For the size inlet tubes considered for the Maxi-Therm units (7.62 - 10.16 cm), the use of corner radii as small as 1.14 - 1.52 cm will reduce the entrance pressure drop by a factor of $10^{(2)}$. Since tank connections will be supplied with each Maxi-Therm unit, the purchaser will not be required to obtain a special exit assembly.

2.2.1.3 Tank Entrance Pressure Losses

Unlike the liquid flow at the tank exit (Section 2.2.1.2), the losses at the liquid entrance to the tank occur regardless of the shape of the tank connection. These losses are quite substantial, equaling one velocity pressure head ($\frac{1}{2} \cdot V^2/g_c$) in magnitude. Therefore, from Bernoulli's equation, the static pressure in the tube at the point of entrance to the tank is just equal to the tank total or static pressure ($V_{TANK} \approx 0$). To minimize this loss, then, it is necessary to reconvert as much of the velocity pressure head to static pressure head as possible before reaching the tank entrance. In addition, it is necessary to perform this operation with as low a loss as possible such that the overall result will be minimal total loss.

The solution to this problem is a diffuser or cone-shaped tank connection which permits a decrease in velocity to occur, in the direction of flow, without creating unwanted eddy conditions. To determine the proper diffuser size, it is necessary to simultaneously consider both the pressure drop of the pipe-diffuser interface as well as the pressure at the tank entrance.

From Reference 2, the pressure resistance coefficient in going from the pipe to the diffuser is given, in terms of the pipe diameter, by

$$K_1 = 0.8 \cdot \sin\left(\frac{\theta}{2}\right) \cdot (1 - \beta^2) \quad (\theta \leq 45^\circ) \quad (2-5)$$

where

K_1 = resistance coefficient at pipe-diffuser interface

θ = total cone angle; radians

$\beta = d_1/d_2$ = diameter ratio

d_1 = pipe inside diameter (diffuser inlet diameter), cm

d_2 = diffuser outlet diameter (tank inlet diameter), cm

and the entrance loss coefficient at the tank is given by

$$K_2 = 1 \quad (2-6)$$

The overall pressure drop of the diffuser, in the absence of wall friction, is then

$$\Delta P_1 = \frac{V_1^2}{2g_c} \left(0.8 \cdot \sin \left(\frac{\theta}{2} \right) \cdot (1 - \beta^2) \right) + \frac{V_2^2}{2g_c} (1) \quad (\text{dynes/cm}^2) \quad (2-7)$$

where

ΔP_1 = total pressure drop with diffuser, dynes/cm²

V_1 = liquid velocity at d_1 , cm/sec

V_2 = liquid velocity at d_2 , cm/sec

and the other terms are as previously defined.

In contrast, the pressure drop with a straight entrance of diameter d_1 is

$$\Delta P_2 = \frac{V_1^2}{2g_c} (1) \quad (\text{dynes/cm}^2) \quad (2-8)$$

The ratio of ΔP_1 to ΔP_2 is given, following simplification, by

$$\frac{\Delta P_1}{\Delta P_2} = 0.8 \cdot \sin\left(\frac{\theta}{2}\right) \cdot (1 - \beta^2) + \beta^4 \quad (2-9)$$

In Figure 2.4, representative values of $\Delta P_1/\Delta P_2$ are given for pipe sizes relevant to the Maxi-Therm system. For the preferred piping size (7.62 cm I.D.), the curves indicate that diffuser outlet diameters of 12.7 cm (5 in.) provide an adequate reduction in pressure, for 15.24 cm long units, with a minimum occurring near 13.97 cm (5.5 in.). This curve does, in fact, show a rise at 15.24 cm (6 in.) owing to the large cone angle and its associated pressure drops. For both sets of curves, the use of longer diffusers indicates a further decrease in pressure drop, but not of sufficient magnitude to warrant the additional size and cost.

2.2.1.4 Pressure Drops in Heat Transfer Elements

The heat transfer tube elements in the parallel flow exchanger display three pressure drop characteristics found in other portions of the system, these being entrance, exit, and straight-pipe viscous flow pressure losses. In the parallel-series system, an additional pressure loss occurs in the 180° return bends. The second of these pressure losses, occurring at the exit, can only be decreased by utilizing a diffuser section, as discussed in Section 2.2.1.3. This is not a practical approach for each tube in the exchanger module due to difficulties in producing such a configuration, and thus exit pressure loss cannot be decreased.

Viscous flow losses within the tube elements can be represented by equation 2-3, with the length and diameter of the tube once more being the variables of interest. Both, however, are intimately tied to the heat transfer characteristics of the exchanger, and can only be considered in a total system analysis. Changes in performance with a variation in these parameters are thus delayed until Section 2.2.4.

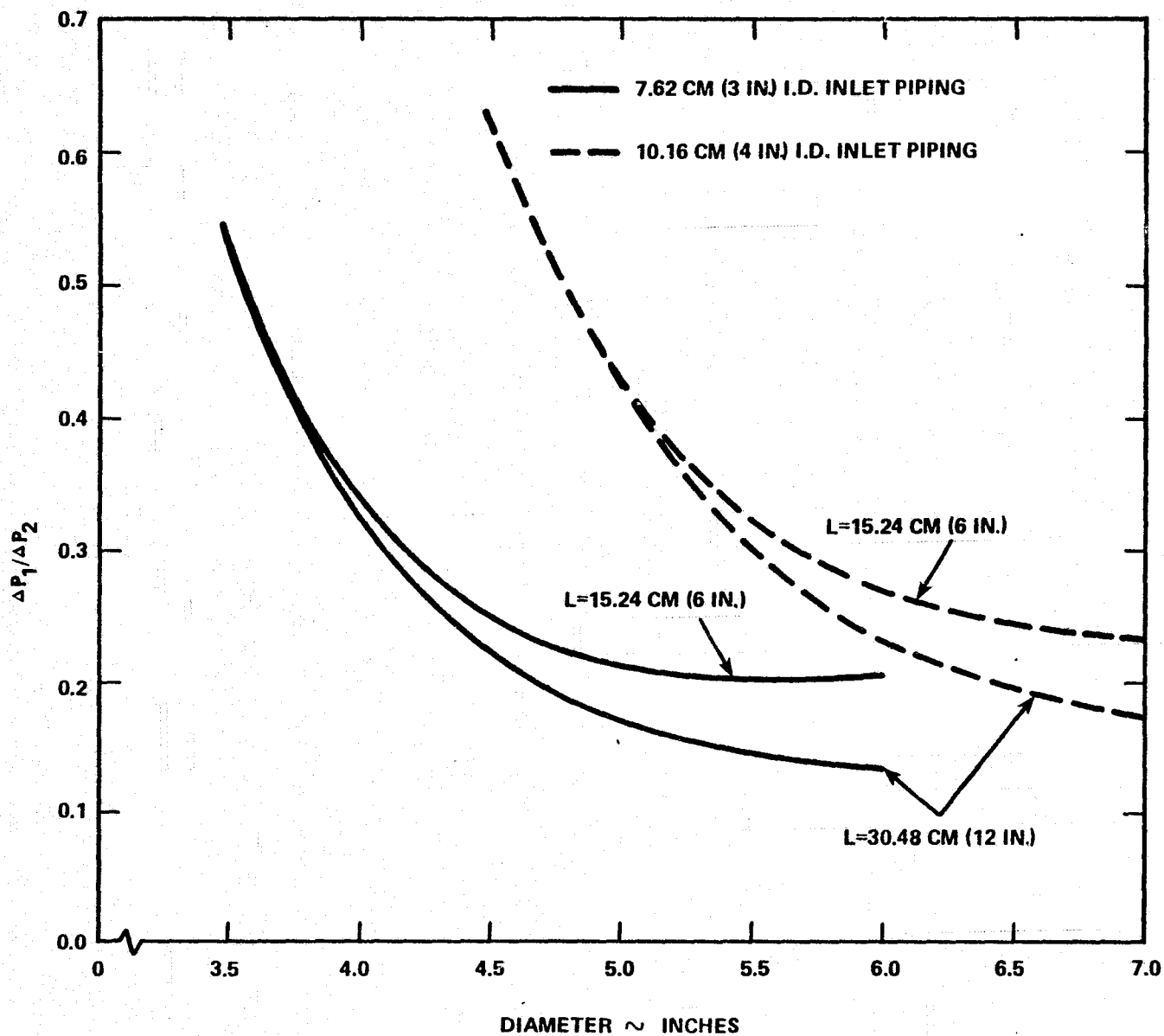


FIGURE 2.4. VARIATION IN THE RATIO OF TANK INLET PRESSURE DROP WITH DIFFUSER (ΔP_1) TO TANK INLET PRESSURE DROP WITHOUT DIFFUSER (ΔP_2). L IS THE DIFFUSER LENGTH. CONVERSION FACTOR FOR DIAMETER - 1 INCH = 2.54 CM.

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The third pressure drop encountered in regard to the heat transfer elements involves the tube entrance losses. Once again, as discussed in Section 2.2.1.2, a modest radius of curvature at the opening will eliminate the formation of a vena contracta and the associated pressure loss. By subjecting the tube sheet or inlet header to a forming operation in which the tube openings are rounded and extended from the plane of the basic flat stock (parallel system) or tube (parallel-series system), this curvature can be provided. An additional benefit of this operation is the production of a larger area over which a solder or braze joint can be formed, thus adding durability to the exchanger joints. Unlike automotive radiators, it will be necessary to orient the extensions toward the tube elements at the inlet to achieve the rounded configuration.

The 180° return bends found in the parallel-series arrangement create pressure drops in a manner similar to that of 90° elbows, although of greater magnitude. For modeling purposes, this pressure drop is taken to be twice that of the Equation 2-4, although it is recognized that in practice, multipliers somewhat less than two are generally present.

2.2.2 Experimental Tests on Liquid Side Heat Transfer

Early estimates of performance for the horizontal thermosyphon indicated a need to operate the exchanger module, over much of its heating range, in a turbulent liquid flow condition. Although pressure drop is increased by operating in this mode, as opposed to laminar flow, compactness in size is only possible by creating an efficient transfer of heat between the water and tube wall. In addition, to best utilize the external fin densities possible in modern liquid-to-air exchangers, some level of parity must be obtained between the water-to-tube wall conductance and the tube wall-to-air conductance.

To provide data on turbulent heat transfer in tubes of the size range under consideration, and to permit additional knowledge to be obtained on the general characteristics of horizontal thermosyphons, a laboratory-scale test facility was constructed. This unit, shown in Figure 2.5, consists of a vertical tank 1 meter high filled with a submersible pancake heater,

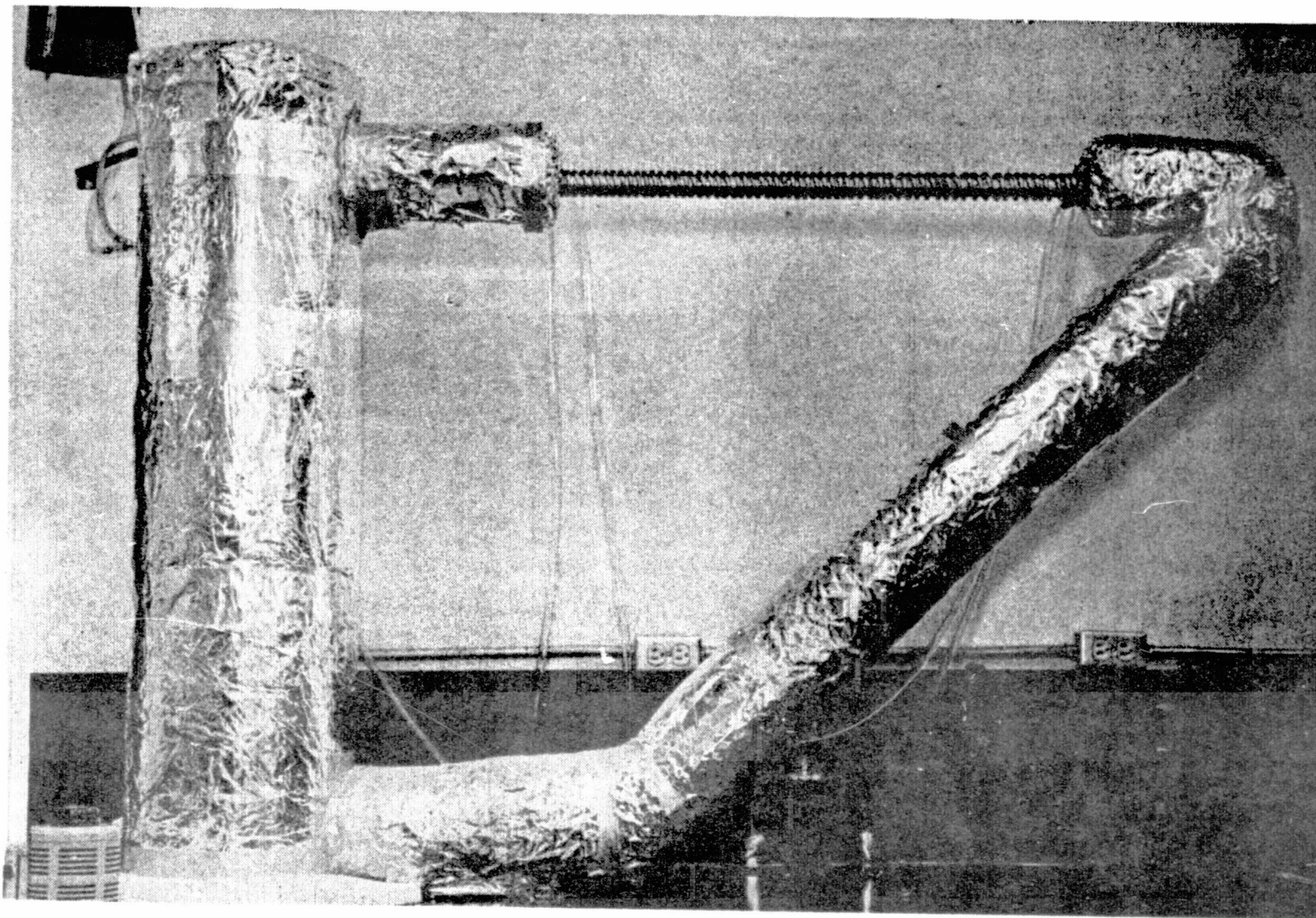


FIGURE 2.5. LABORATORY TEST UNIT FOR EVALUATING PERFORMANCE OF HORIZONTAL THERMOSYPHONS. TEST SECTION SHOWN IS 2.68 CM I.D. UNIT WITH DUAL WATER COOLING TUBES.

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located near the bottom of the tank. The heat transfer test section used in experiments consisted of a 2.68 cm inside diameter copper tube wrapped with twin water coolant lines. The connections to these lines were made in such a way that one coolant stream flows countercurrent to the test stream while the other flows concurrent.

Connection of the test section to the tank was accomplished using 3.95 cm diameter tubing having a total length of 2.2 meters. Heat loss from this transfer piping, as well as from the vertical tank, was minimized through the use of polyurethane foam insulation. The thickness of this insulation varied from 7.62 cm to 10.2 cm depending on location. Power for the test runs was supplied with a 240 volt variac, and all temperature measurements were accomplished using type KK thermocouple wire in conjunction with a Doric DS-350 meter (0.1°C resolution).

All tests were performed under steady state conditions, with measured (Fluke Mod. 8000A) values of input power being recorded along with system temperatures. The temperature data was then used to define the value of various fluid properties, with the input power and temperature drop across the loop being used to calculate mass flow rate. Data reduced in this manner was then studied to determine the best correlation method, the result of which is shown in Figure 2.6.

In this figure, the ratio of $Nu/\sqrt{Pr}$ is plotted as a function of Reynolds number, a correlation technique suggested by the results of Reference 3 and the empirical equations presented by Kays.⁽⁴⁾ Along with the data is plotted this empirical relation from Kays, the agreement being quite satisfactory. The curve does, in fact, fit the data well into the Reynolds number range below 2100, an area generally considered to be characteristic of laminar flow. A possible explanation for the lack of a jump, or rapid increase, in the Nusselt number above a Reynolds number of 2100 is the occurrence of secondary, free convection effects impressed on the axial flow below this value. The existence of this phenomena, and its effect on heat transfer is a well-documented fact (5, 6, 7, 8), although an accepted empirical correlation is not presently available. In general,

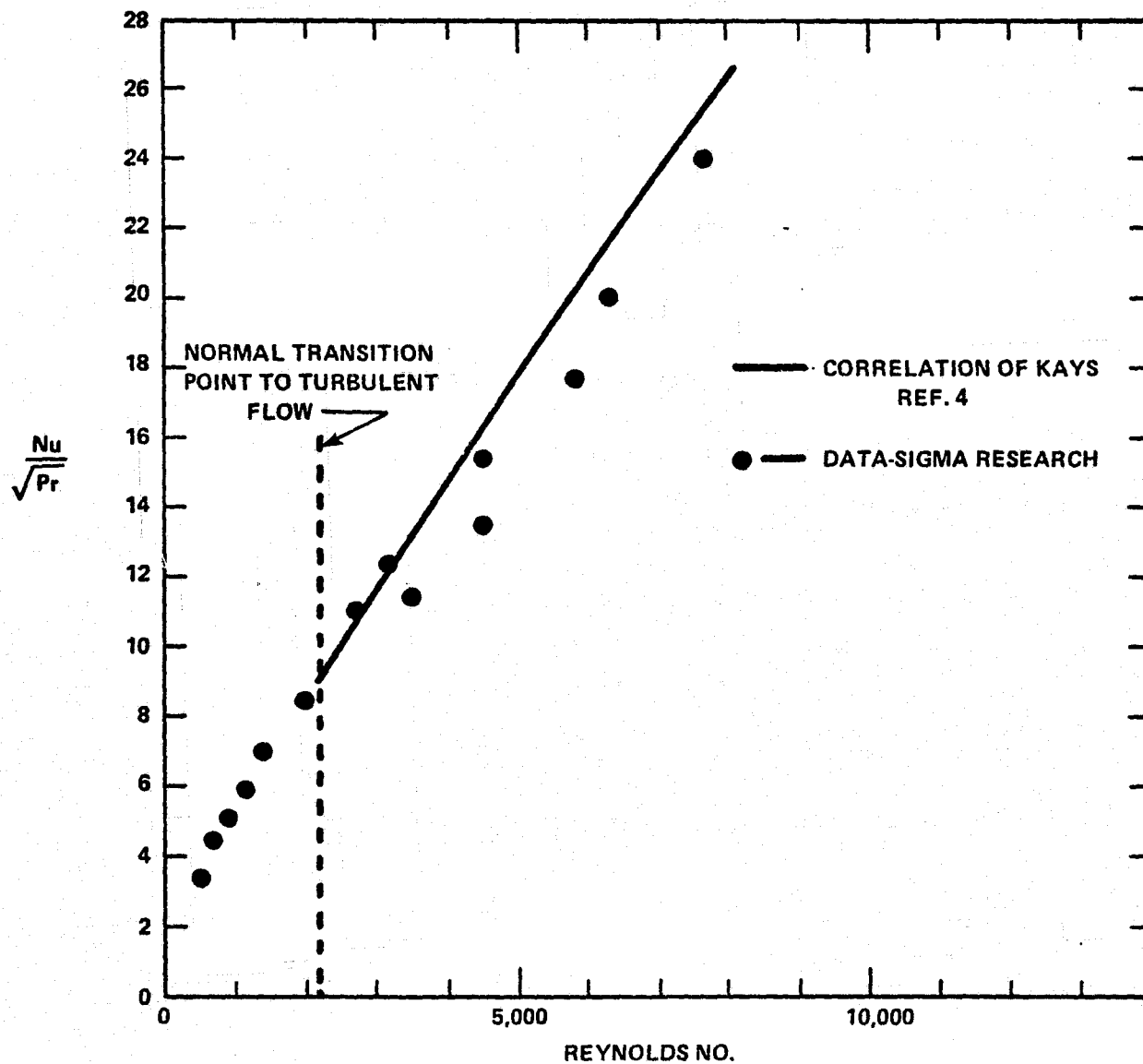


FIGURE 2.6. EXPERIMENTAL DATA FROM HEAT TRANSFER TESTS WITH HORIZONTAL THERMOSYPHON LABORATORY TEST UNIT. THE CORRELATION OF KAYS IS INCLUDED FOR COMPARISON. WATER WAS USED AS THE WORKING FLUID IN ALL TESTS, AND THE TEST SECTION WAS 2.68 CM. I.D.

the magnitude of the Nusselt numbers resulting from these studies is not unlike those seen in turbulent flow, so the characteristics seen in this experiment are realistic.

2.2.3 Air Side Heat Transfer and Pressure Drop

On the air side of the exchanger, efficient coupling to such a dilute fluid requires the use of fins or extended surfaces. In general, a compact design and a minimum cost per unit output can be achieved with fin-to-bare tube area ratios of from 10 to 15. In this range of packing density the Nusselt number for multi-row units is given, according to (9), by

$$Nu = 0.134 Re^{0.681} Pr^{0.33} \cdot \left(\frac{s}{\ell}\right)^{0.2} \cdot \left(\frac{s}{t}\right)^{0.1134} \quad (2-10)$$

where

- Nu = Nusselt number, hD/K
- Re = Reynolds number, GD/μ
- Pr = Prandtl number, $\mu C_p/K$
- s = axial fin-to-fin inner spacing, cm
- ℓ = fin length, cm
- t = fin thickness, cm
- h = convective coefficient, $W/cm^2-^{\circ}C$
- D = tube diameter, cm
- K = air thermal conductivity, $W/cm-^{\circ}C$
- G = mass flux at minimum area, $gm/sec-cm^2$
- μ = air viscosity, poise
- C_p = air specific heat, $joules/gm-^{\circ}C$

In general, this relationship has been found to provide very accurate results for individual finned tubes set in a staggered, equilateral triangle arrangement and results of acceptable accuracy for tube and plate fin configurations in similar arrangements.

Pressure drop equations have likewise been developed for finned tubes, with the recommended correlation equation of Robinson and Briggs⁽¹⁰⁾ being given by

$$\Delta P_F = f \frac{G^2 N}{\rho g_c} \quad (\text{dynes/cm}^2) \quad (2-11)$$

where

ΔP_F = pressure drop across tube bundle, dynes/cm²

G = mass flux at minimum area, gm/sec-cm²

N = number of tube rows in direction of air flow

ρ = air density, gm/cc

g_c = conversion factor = 1

and

$$f = 18.93 \text{ Re}^{-0.316} \left(\frac{P_T}{D} \right)^{-0.927} \quad (2-12)$$

where

Re = Reynolds number, GD/μ

P_T = centerline-to-centerline spacing ratios of tubes, cm

D = tube diameter, cm

Equation 2-12, it should be noted, applies only to tube bundles arranged in a staggered, equilateral triangle pattern. Although developed for individual

finned tubes, Equation 2-12 can be used to calculate plate fin-tube pressure drops. In general, pressure drops calculated for these configurations are somewhat higher ($\approx 15\%$) than measured values.

2.2.4 Parametric Evaluation of System Performance

As noted previously, it was not clear which of the fluid circuiting arrangements, parallel or parallel-series, would represent the best overall heat exchanger design. In addition, a number of other system pressure drop parameters had to be considered along with the basic exchanger module so that the overall system could be evaluated. For these reasons, it was necessary to assembly two computer codes, with one treating the parallel circuited heating system and one treating the parallel-series circuited system. In Appendix A, listings for both of these computer codes are provided.

Basically, the codes were used to perform parametric studies on the key system parameters, particularly the tube diameter and fin arrangement of the heat transfer elements and the diameter of the inlet and outlet piping. In all cases, the length of the heat transfer elements was set at 0.61 meters. An increased length, it was felt, would be detrimental to consumer acceptance in that a longer unit would present more difficulties in siting. Restrictions were also put on the width, which was also 0.61 meters, for similar reasons. Maximum inlet and outlet piping diameter was set at 10.2 cm since the ready availability and aesthetics of a larger pipe diameter are considerably decreased.

Additional system variables which were fixed for this study included the inlet air temperature (21.1°C - 70°F), the tank temperature (60°C - 140°F), the fin material (0.0203 cm aluminum), and the characteristic thermosyphon driving pressure height (122 cm - 4 ft).

In Figure 2.7, the results are shown for the first parametric series which dealt with the variation in performance due to changes in the tube diameter of the heat transfer elements for parallel circuited systems. As indicated

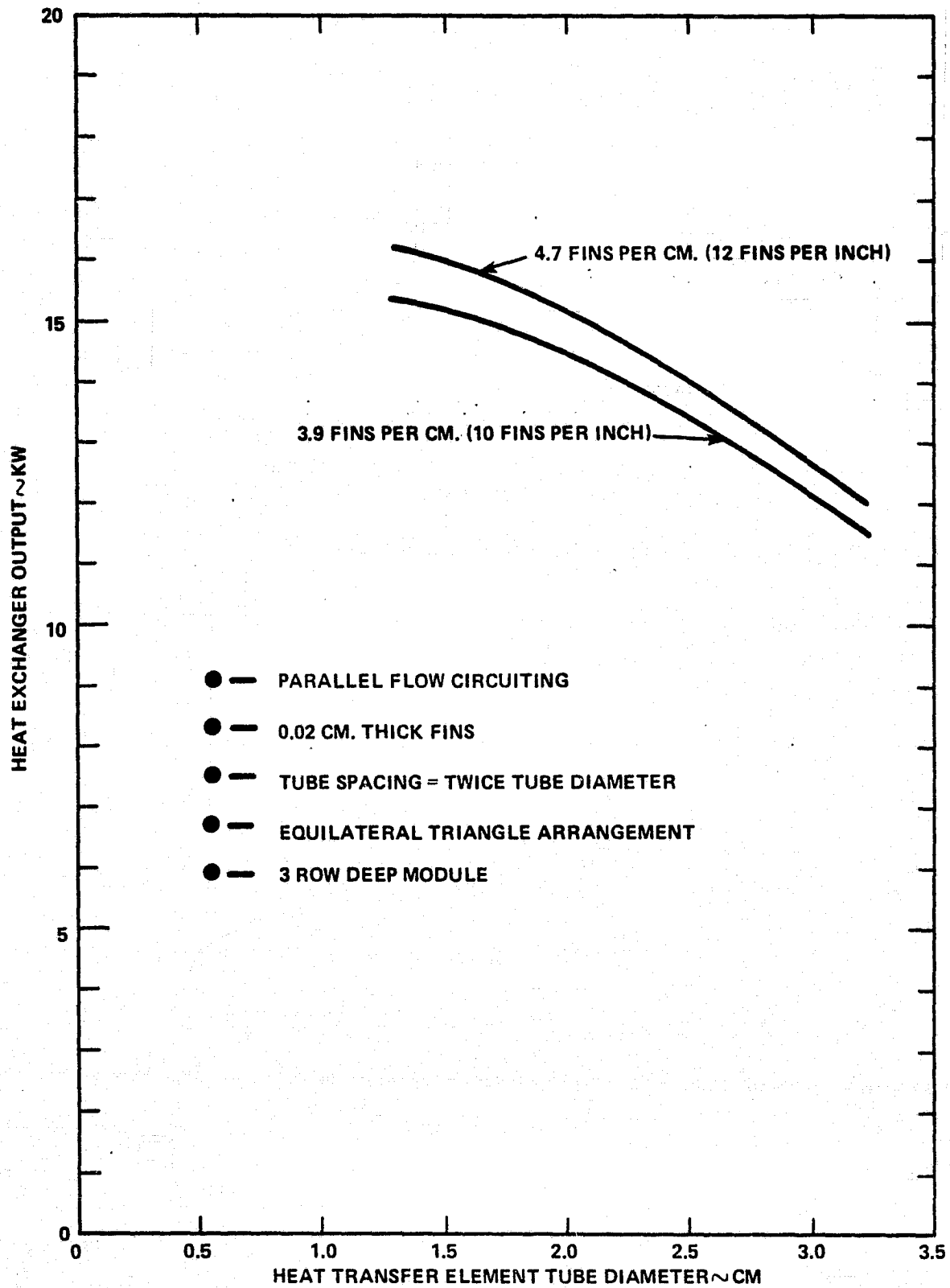


FIGURE 2.7. BEHAVIOR OF PARALLEL CIRCUITED HEAT EXCHANGERS WITH CHANGES IN TUBE DIAMETER AND AIR SIDE FIN DENSITY.

by the curves, an increase in diameter above 1.27 cm (0.5 inch) results in continually decreasing performance, primarily due to water-side heat transfer effects. Although the product of the convective coefficient, H , and the individual tube area, A , which is the element conductance, continues to increase, this rise is not sufficient to overcome the effect of a decreasing number of tube elements. This is primarily due to the transfer system pressure drops, which become more dominant as the diameter of the tube elements increases. An increase in the transfer system tube diameter from the 7.62 cm value used in preparing this figure to 10.16 cm would alter these, with the primary effect being a shift to the right or toward larger tube diameters. It should be noted that, where an even number of tubes did not yield the maximum 0.61 meter width, the next lower integer number of tubes was utilized.

A second effect demonstrated in Figure 2.7 concerns the use of higher air-side fin densities. Considering the rather modest increase in densities in the figure, the change in performance is quite pronounced. Part of this change is the result of more efficient air-side coupling, while a portion is also due to improved water-side coupling. This coupled effect comes about due to the thermosyphon water flow, wherein a decreased water exit temperature from the exchanger promotes a larger driving pressure and consequently higher liquid mass flow rates. This higher mass flow rate then acts to increase the water side convective coefficient.

In Figure 2.8, the performance is once again shown for the parallel circuited exchanger of Figure 2.7 (3.9 fins per cm) along with a similarly sized parallel-series circuited unit. As shown, the parallel-series configuration has a much flatter performance curve with varying tube diameter. This is due to the fact that the primary pressure drop now occurs across the heat exchanger. As the tube diameter of the exchanger elements increases, water side conductance now increases at a high enough rate to compensate for the reduced number of elements. An optimum, for the overall system shown, occurs near 2.1 cm (0.83 inches), although variations of ± 0.5 cm decrease the performance by less than 1.5%.

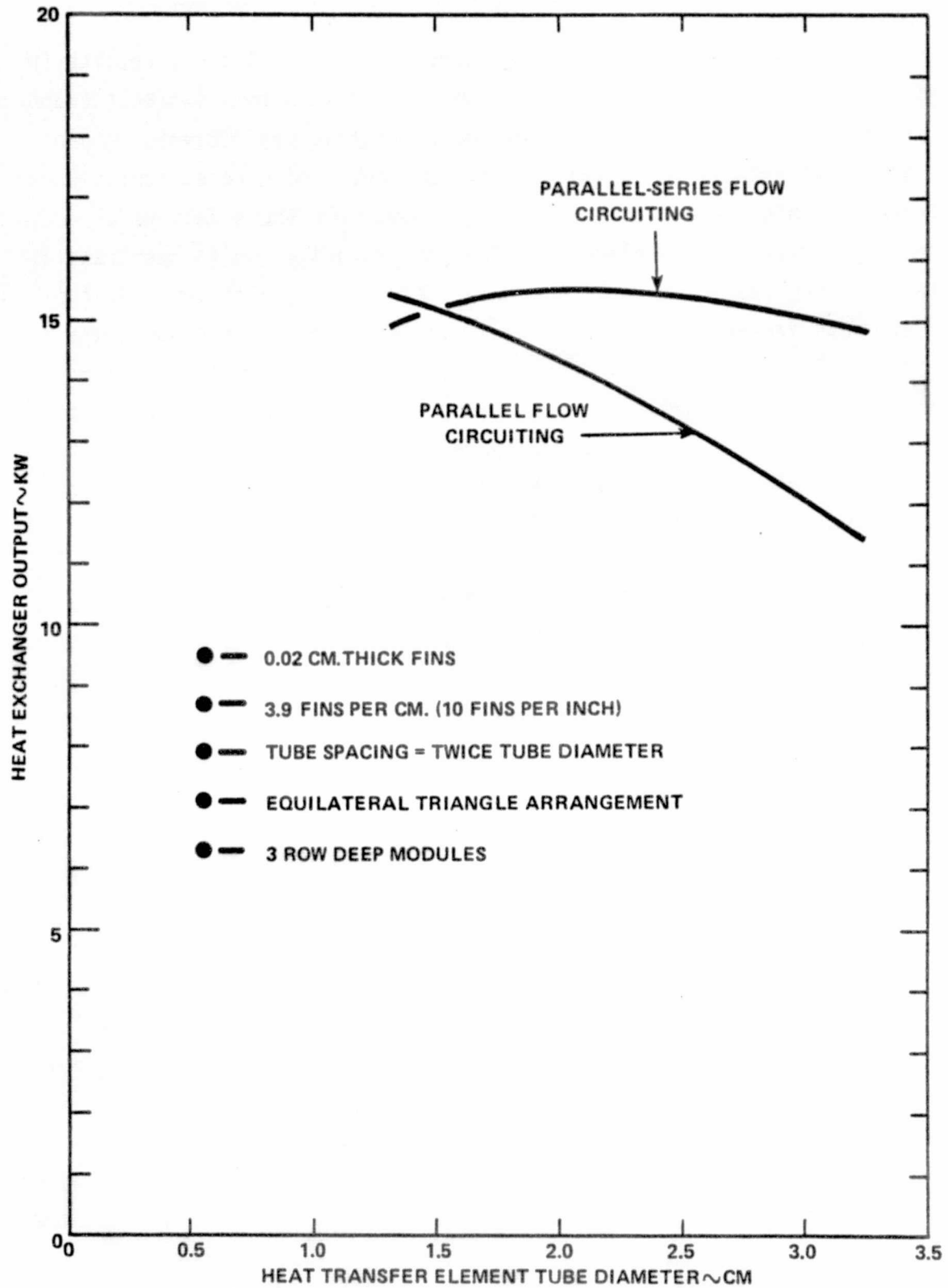


FIGURE 2.8. COMPARISON OF HEATING PERFORMANCE IN PARALLEL AND PARALLEL-SERIES CIRCUITED HEAT EXCHANGERS.

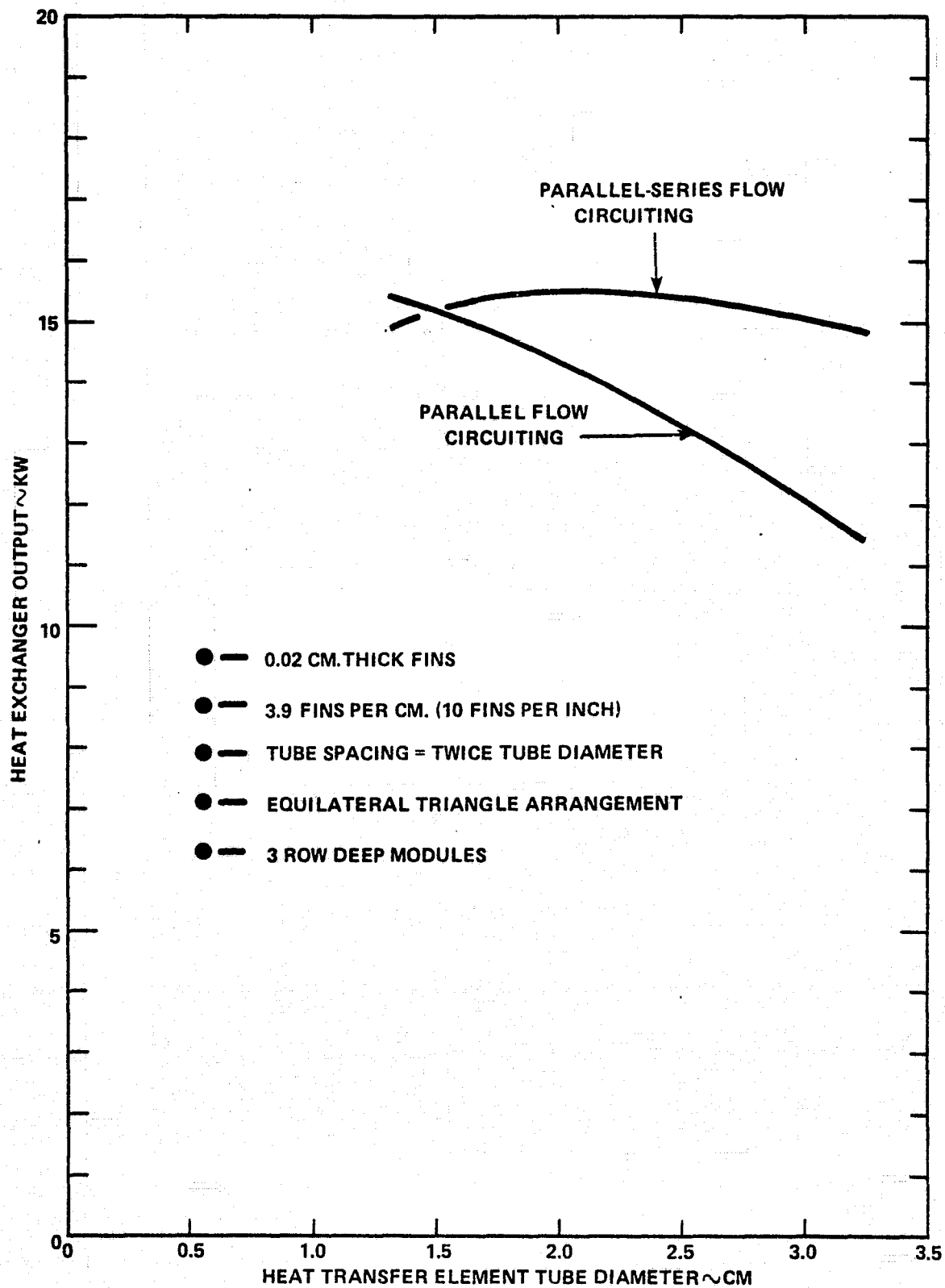


FIGURE 2.8. COMPARISON OF HEATING PERFORMANCE IN PARALLEL AND PARALLEL-SERIES CIRCUITED HEAT EXCHANGERS.

The fact that the predominant pressure drop occurs across the heat exchanger, rather than across the transfer system piping, permits more variation in the transfer system piping, as discussed previously. Also, the relative independence of performance on element tube diameter is an important feature in design since it permits a good deal of flexibility in the selection of components. In particular, the tube diameters indicated as optimum or near optimum sizes are within the range of those found in commercial air cooled heat exchangers composed of round tubes and flat, continuous fins. By utilizing this type of design, as opposed to the spiral wound, individual fin tube design, increased durability and decreased cost can be realized in production units. Although somewhat different in shape, the two designs are quite similar in performance and behavior, as noted previously in the discussions on modeling equations (Section 2.2.3).

The possibility of using such a design led to further analytic investigations of available core configurations, with the unit closest to optimum (from Figure 2.8) being composed of 1.59 cm O.D. tubes with a 3.81 cm tube spacing. Since constraints had previously been placed on exchanger length and width, the primary area to be investigated concerned the exchanger depth, or more specifically, the number of series connected rows in one of the parallel paths. From an initial parametric study, it was found that module width could be decreased below the 0.61 meter value while still attaining the desired performance (≈ 16 kW), so further parametric studies were performed. In Figure 2.9, the results of two of these parametric studies are shown. As indicated, units having a width of either 0.46 or 0.53 meters provide the desired level of performance when 4 row deep configurations are utilized. The use of 4 row deep units, as indicated by the curves, represents a good economic choice, since the output per row of depth has not begun to decline as rapidly as in modules of 5 rows or more in depth.

From these final parametric studies, the 0.53 meter wide module was chosen for development as the prototype unit. Discussions with vendors are now proceeding so that final specifications can be prepared and a source for the prototype unit can be selected.

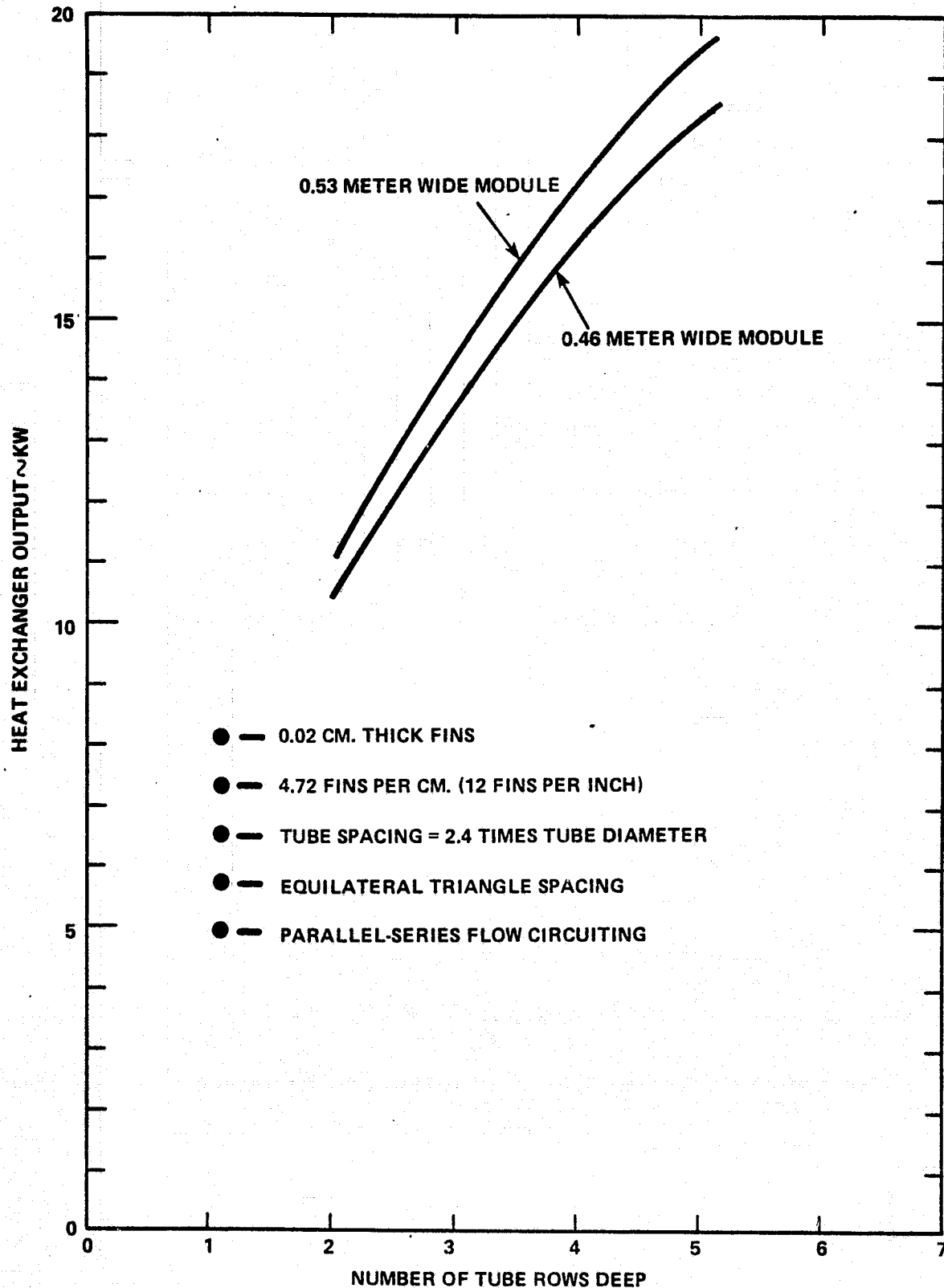


FIGURE 2.9. BEHAVIOR OF PARALLEL-SERIES CIRCUITED HEAT EXCHANGERS WITH AN INCREASING NUMBER OF ROWS DEEP.

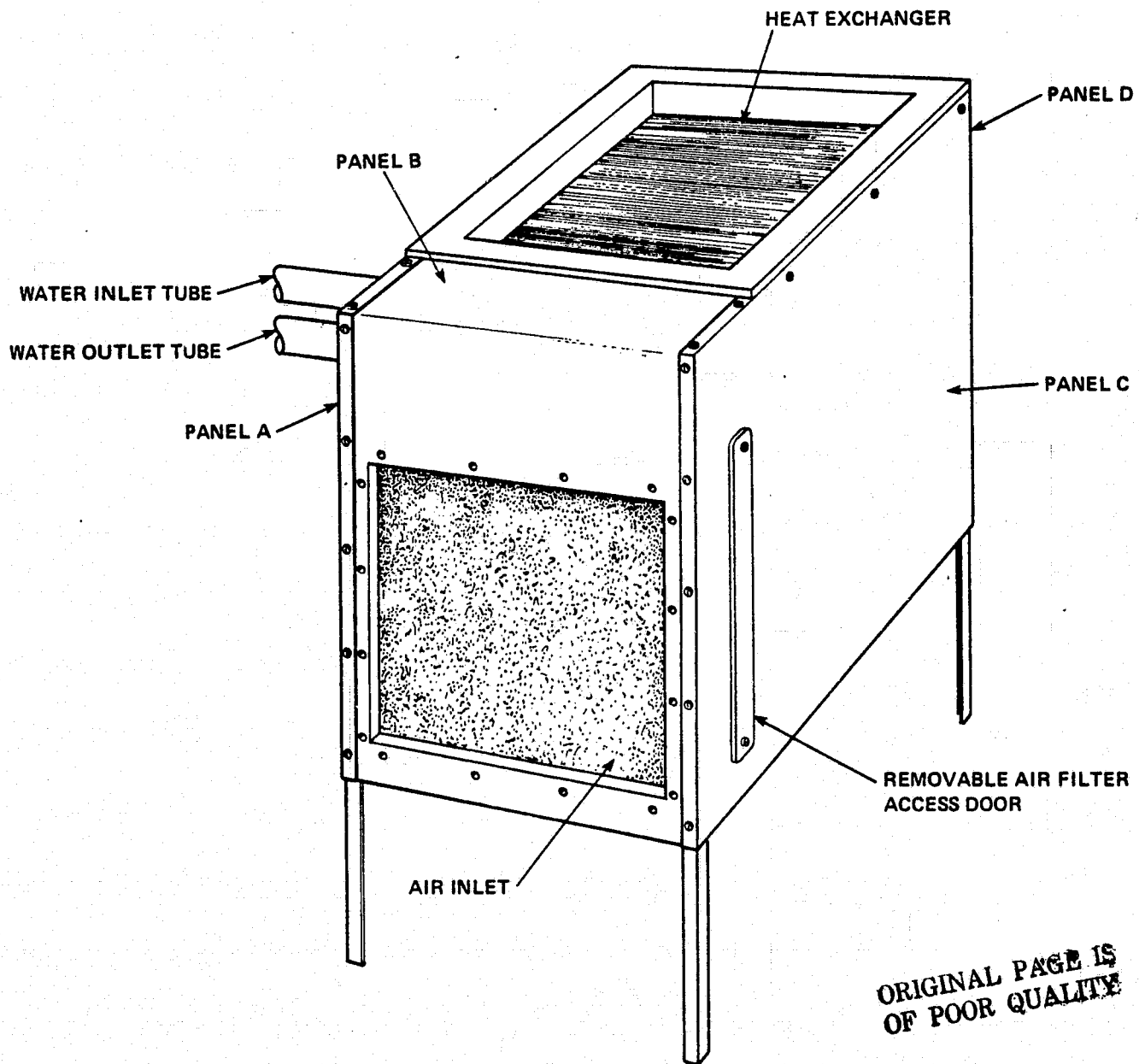
3.0 MAXI-THERM CABINET DESIGN

The primary purpose of the cabinetry for the Maxi-Therm is to provide a pleasing appearance to potential buyers. At the same time, however, this exterior shell must be designed to permit easy module maintenance, low cost production, and simple assembly and disassembly. In addition, where the exterior shell can be designed to perform other functions, such as acting as an air plenum, these features should be incorporated.

In Figure 3.1, a conceptual view is shown of the Maxi-Therm S-101 module with the exterior shell in place. The shell consists basically of four sections, labeled Panels A, B, C, and D in Figure 3.1. The general order of assembly for the unit will consist of first attaching Panels B and D to the frame, followed by the installation of the heat exchanger assembly. After attaching the heat exchanger to the frame and Panels B and D, Panels A and C can be slipped over the outer edges of B and D and attached to the frame.

By designing the shell in this manner, a number of advantages become possible. First, in the production of the shell, the formation of complicated or intricate bends is not required, thus permitting the components to be manufactured on a normal brake and shear. In addition, the use of Panels A and C to form the finished corners permits Panels B and D to have flat, as-produced edges and further decreases the number of critical dimensions required in their production.

Combining the function of the cabinet with other requirements is also possible with this design. As shown, a portion of Panel B acts as the air inlet and to its backside at this position is attached the filter holder. Access to this filter is provided by the indicated removable cover on Panel C. The cabinet as a whole further acts as both the inlet and outlet air plenums, with a horizontal, sheet steel divider separating the two sections. The seal between the inlet and outlet plenum is provided by foam rubber strips which are bonded to horizontal frame stiffeners positioned directly below the divider and extending the entire circumference of the unit.



7703-948.12

FIGURE 3.1. CONCEPTUAL VIEW OF PROTOTYPE MAXI-THERM S-101 WITH CABINET INSTALLED.

Scheduled maintenance for the unit will consist only of changing the air filter, a task easily performed by removing the indicated cover and sliding out the old filter. When additional attention is necessary, such as repairing the blower or electric motor, access can be gained by removing Panel C. This operation, it should be noted, does not require removal of either the water inlet or outlet connection, nor the blower motor electrical supply, the latter of which enters through Panel D. Should repairs requiring removal of the heat exchanger become necessary, Panel A must also be removed.

Originally, the entire electrical component package was also considered for mounting within the shell, although this design was modified to position the electrical components within a separate electrical box attached to the outside of Panel D. Primarily, this was done to eliminate possible conflicts with local electrical codes concerning the danger of 115 VAC junctions in the proximity of a heat exchanger containing water.

4.0 QUALITY ASSURANCE PROGRAM

4.1 QA Plan Structure

The goal of any quality assurance program is to provide the customer with a product which is free of defects and capable of meeting the required operating specifications. In the QA plan designed by Sigma Research for the Maxi-Therm Heating Modules, this has been a central theme, with early inspection and review of components helping to make the Quality Assurance and production phases compatible. To serve as a guide and outline for this program, a flowchart has been prepared and is presented in Figure 4.1.

As indicated in Figure 4.1, drawings from the designer are reviewed by engineering, and if satisfactory, work orders are formulated per drawing requirements. Materials and components purchased or fabricated are inspected by Receiving. If satisfactory, materials are placed in Materials Inventory. If not to specification, the components are sent to the Material Review Board (MRB) for disposition. The MRB has several avenues of action. If it is felt that the component can be reworked, it is reworked and reinspected. If the component cannot be reworked, it is scrapped. If the component is out of specification because of vendor error, the part is returned to the vendor with a Notice of Rejection. If the number of rejected parts can be reduced by an appropriate design change with no sacrifice in component function, then a drawing change order is initiated.

Within the manufacturing process, quality is maintained through the use of fabrication/assembly outlines and by work instructions and manufacturing process inspection criteria.

If during the manufacturing process a part or component does not meet inspection criteria, a Failure and Rejection Report (FARR) is written, and disposition of the part is subject to action by the MRB.

The finished article must pass Final Acceptance Tests prior to packaging. The packaging process is documented with a Packaging, Packing, and Markings Inspection Form.

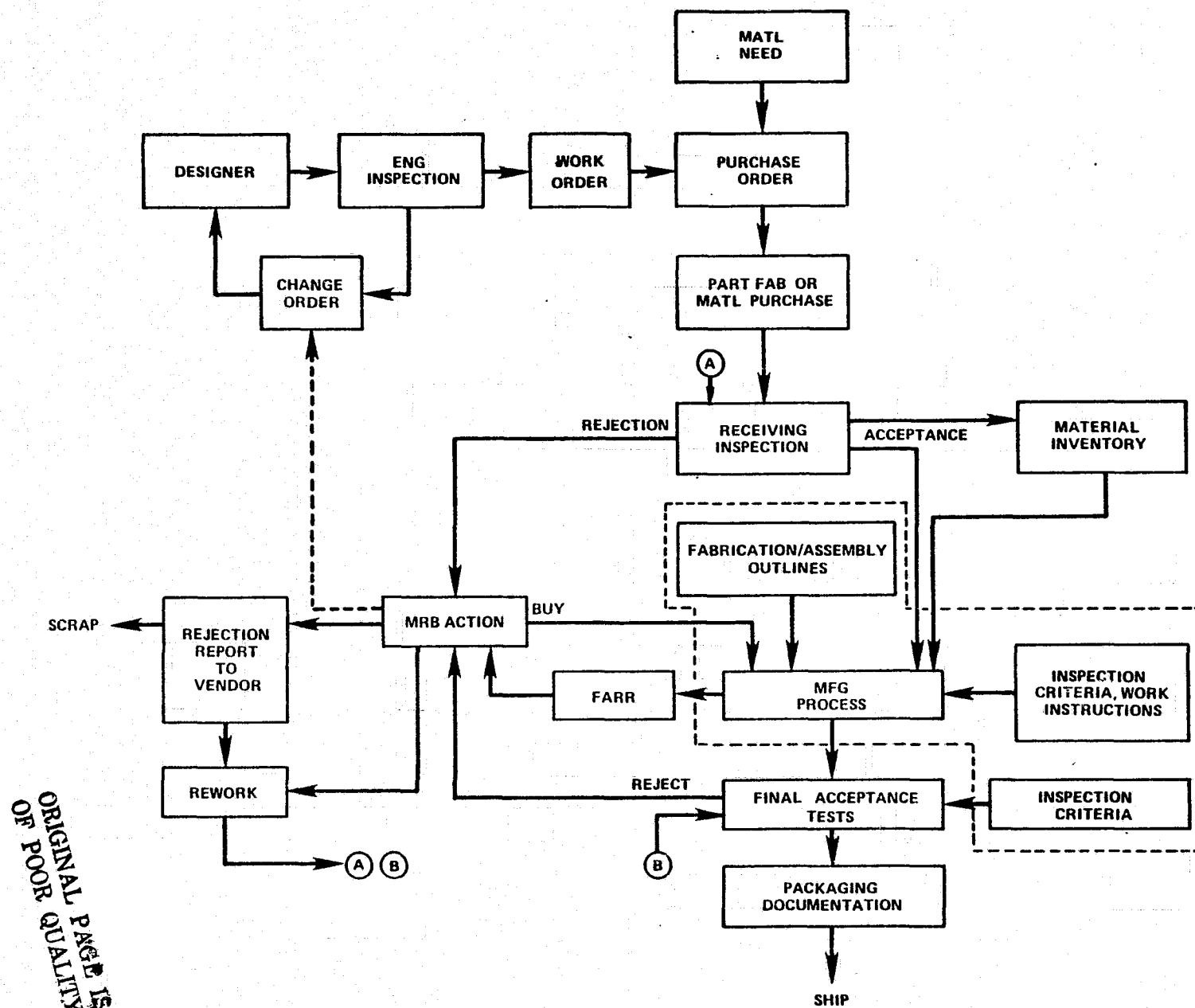


FIGURE 4.1. QUALITY ASSURANCE FLOW CHART

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4.2 Quality Assurance Forms

The Quality Assurance Forms have been deleted from this report.

5.0 SHUT-OFF VALVE SELECTION

During the early development of a thermosyphon water-to-air heat exchanger, it became apparent that the unit possessed one particular trait not usually found in heat exchangers, that being the requirement for a positive shut-off valve. In normal forced flow exchangers, cessation of water pump operation is sufficient to halt exchanger operation. In thermosyphon units, however, the movement of air through the exchanger and the subsequent decrease in water temperature creates the liquid driving force which in turn causes the liquid circulation. To shut the device off then, either air movement or liquid circulation must be eliminated.

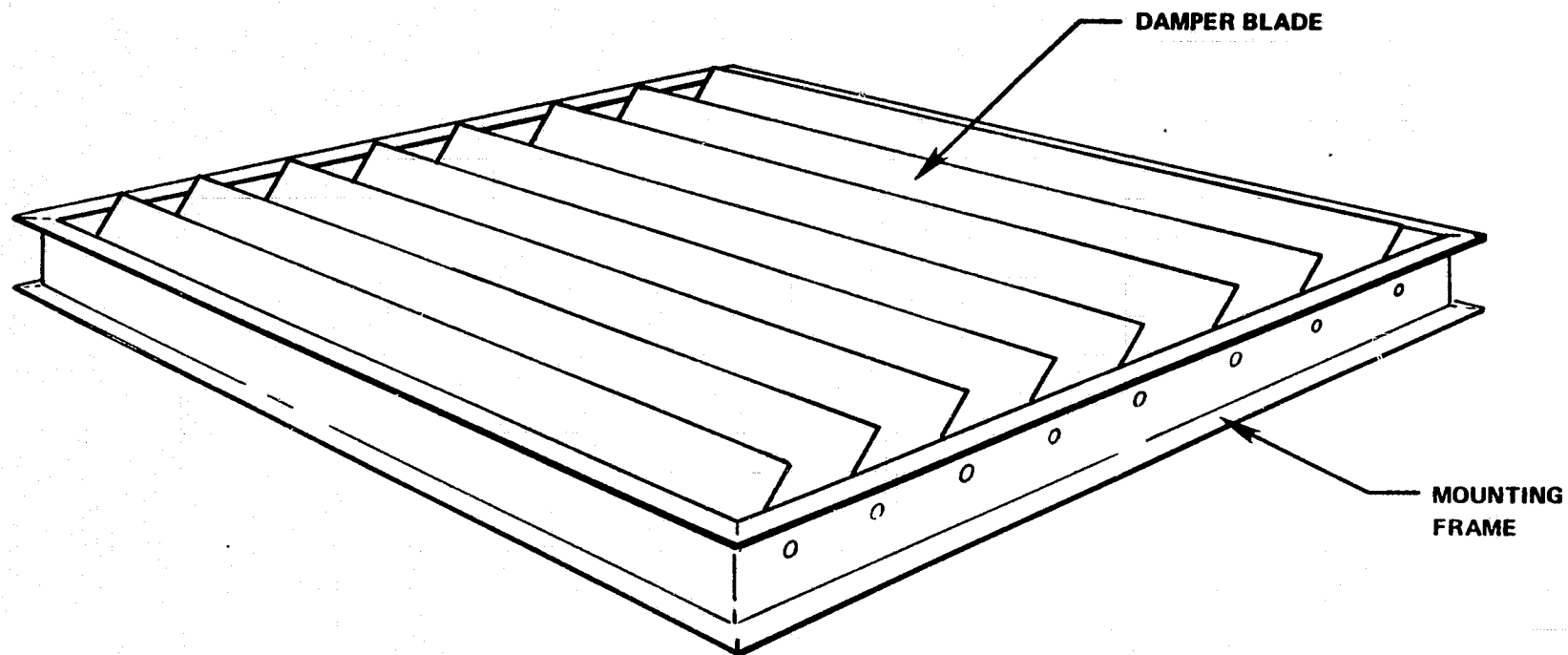
Initially, investigations were made into the use of a water-side shut-off. This type of device, it was found, would require use of an electric solenoid, a component which added both cost and a potential area for maintenance/repair in the future. In addition, most types considered required a movable shaft and sealed penetration through some portion of the transfer system piping. Besides the cost, this represented a potential area for leakage and additional maintenance. Another consideration was the possible effect on transfer system pressure drop that a valve, even in the full open position, would create and the corresponding effect that it would have on module thermal performance.

When it was found that a water-side shut-off would not be desirable, investigations were commenced to define possible air-side shut-off valves. Since the position of the valve with respect to the heat exchanger has a strong bearing on the design, this was the first area to be investigated. Consideration was given to two locations during these studies, one being the outlet ducting and the other, the plenum area immediately above the heat exchanger. Since the outlet ducting will not be included with the Maxi-Therm unit, the installation of a factory supplied shut-off in the outlet ducting would entail additional cost to the homeowner and would not guarantee proper installation. In addition, insulation would be required on the section of ducting between the exchanger and shut-off valve to avoid excessive heat loss. These drawbacks were felt to be serious enough to consider the second location, which is within the outlet plenum. Although this position does require

fitting the valve to a predetermined opening size, thus eliminating flexibility in sizing, it does eliminate the need for additional insulation and allows the valve to be installed at the factory.

Following selection of the latter position for the valve location, attention was next turned to the design of the component. For this application, a multi-vane damper similar to that shown in Figure 5.1 appeared to be a good choice, with the method of actuation being the only area of investigation. Basically, this actuation can be provided by either a motor or differential pressure across the damper, such as that generated by the blower fan. The latter of these two was the preferred choice, since the cost of a motor and the associated linkage generally raises the price by \$50-\$100. In addition, the passive opening system does not require maintenance or adjustment after leaving the factory.

For the prototype system, a further decision was made, that being to purchase the damper from an outside vendor, if possible. Currently, manufacturers of dampers are being contacted to determine the final design and determine product availability.



5-3

FIGURE 5.1. PASSIVE CONTROL AIR SHUT-OFF DAMPER FOR HORIZONTAL THERMOSYPHON HEAT EXCHANGERS.

7703-948.10

6.0 REFERENCES

1. Schlichting, H., Boundary-Layer Theory, McGraw-Hill Book Co., New York, 1968.
2. Flow of Fluids Through Valves, Fittings, and Pipe, Engineering Division, Crane Co., 1976.
3. Kays, W. M., and Leung, E. Y., International Journal of Heat and Mass Transfer, Vol. 6, 1963.
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5. McComas, S. T., and Eckert, E.R.G., "Combined Free and Forced Convection in a Horizontal Circular Tube," Journal of Heat Transfer, Trans. ASME, Series C, Vol. 88, 1966.
6. Bergles, A. E., and Simonds, R. R., "Combined Forced and Free Convection for Laminar Flow in Horizontal Tubes with Uniform Heat Flux," International Journal of Heat and Mass Transfer, Vol. 14, 1971.
7. Hong, S. W., Morcos, S. M., and Bergles, A. E., "Analytical and Experimental Results for Combined Forced and Free Convection in Horizontal Tubes," presented at the International Heat Transfer Conference, August, 1974, Tokyo, Japan.
8. Depew, C. A., Franklin, J. L., and Ito, C. H., "Combined Free and Forced Convection in Horizontal, Uniformly Heated Tubes," ASME Paper 75-HT-17.
9. Briggs, D. E., and Young, E. H., "Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes," Chem. Engr. Prog., No. 41, Vol. 59.
10. Robinson, K. K., and Briggs, D. E., "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes," A.I.Ch.E. Preprint 20, Eighth National Heat Transfer Conference, August 1965.

7-1

SECOND QUARTERLY REPORT

PROGRESS SUMMARY

Engineering

Review and revision of engineering drawings and documents was initiated to ensure that the action items of the April 27 Prototype Design Review were implemented.

A Qualification Test Procedure has been prepared. This test procedure presents the procedures, records and equipment necessary to verify that the Prototype Passive Heat Exchanger meets the requirements of the Subsystem Performance Specification. This QTP will be submitted for approval in June.

Procurement

The air damper and divided electrical control box were received, which completes receipt of all component parts for the system.

Assembly

The final painting and assembly of the Maxi-Therm passive heat exchanger were completed in May and the Maxi-Therm was connected to the water storage tank to be used during qualification testing. The remainder of the work of test set-up is underway. No problems have been encountered.

Documentation

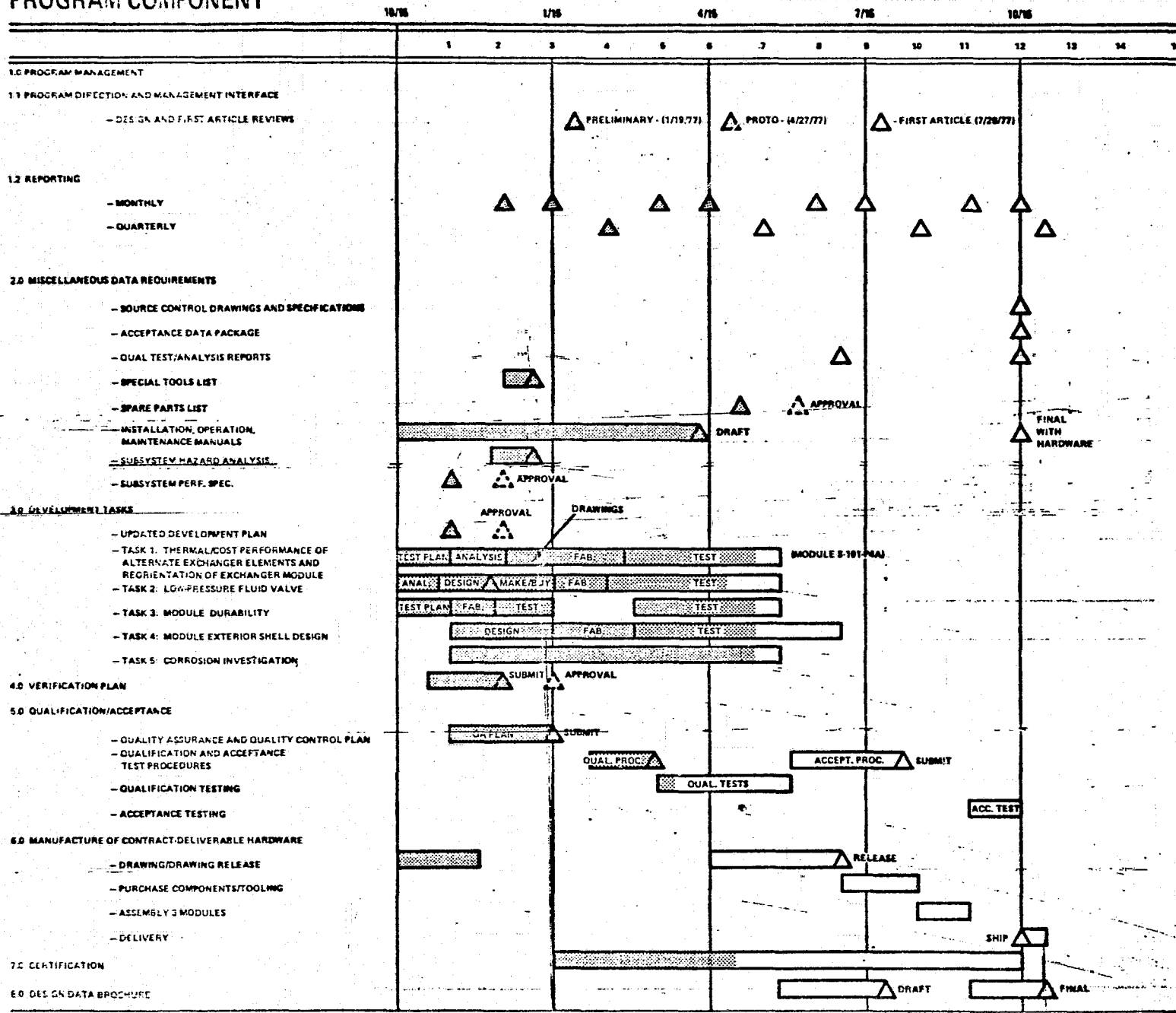
Four type-1 documents were updated and submitted to NASA-MSFC for approval. These include:

1. Subsystem Performance Specification (No. SHC-3068)
2. Verification Plan (No. SHC-3085)
3. Development Plan (No. SHC-3069)
4. Spare Parts List

Schedule

The schedule for the program is shown in Figure 1, with shading () through each scheduled activity to show present status.

PROGRAM COMPONENT



7-3

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FIGURE 1. PROGRAM SCHEDULE FOR SOLAR HEATING MODULE WITH SUPPLEMENTARY HEATING (FOR USE IN SINGLE FAMILY BUILDINGS) (CONTRACT TERM 10/30/76 TO 10/30/77)

7705-141.2

PROGRESS SUMMARY

Design and Development - Prototype System

Further analysis was made regarding the need for corrosion testing of components or parts to investigate the effect of water or ethylene glycol solution in contact with materials of the Maxi-Therm heating module. The final design material selection for tubes and headers is 99.9% copper (Fed. Spec WW-T-799) and the tube-to-header joint is made with an industry standard self-fluxing phos-copper braze compound (92.7% copper, 7.3% phosphorus equal U. S. Navy 47-C-3-Mil-C-20158). Since this metal combination has been used with water systems for many years in domestic, industrial, and naval applications and is an accepted standard practice, there appears to be no useful purpose for a 500-hour corrosion test on the metals.

The assembled prototype system will be inspected before and after qualification testing for any evidence of corrosion or mass transport of materials, which may cause fouling of the internal surfaces of the heat exchanger.

Procurement

Materials received during April included the heat exchanger element, cabinet parts, and air filters. The air damper was delayed but is expected in early May. All parts of the heat exchanger module are now on hand except the damper and the divided electric control mounting box.

Fabrication

The frame and cabinet have been partially assembled and the heat exchanger element set in place to verify measurements and determine the optimum length to cut the heat exchanger header tubes.

Documentation

The required documentation for the Prototype Design Review was completed, including:

1. Special Manufacturing Instructions and Required Materials and parts
2. Installation, Operation, and Maintenance Manual (draft)
3. Spare Parts List
4. Installation Drawings

Program Review

The Prototype Design Review was held at Sigma on April 27, 1977, with Mr. J. B. Hankins of NASA-Marshall.

The program status and documentation was reviewed with the following action items for Sigma:

1. All drawings need to show next assembly.
2. Drawing 101-P4-04 needs more identification on motor and parts.
3. Drawing 101-P4-02 has inlet header missing (that was TBD and has now been firmed up).
4. Recommend monitoring equipment to be mounted by Sigma subject to finalizing at First Article Review.
5. Finalize Type 1 documents for baselining during May.

One action item for NASA-Marshall was:

1. The Contracting Officer should correct the first sub-heading under Article XIII - OPTION of the subject contract document to read as follows:

Heat Exchanger with control valve.

(i.e., this eliminates the words "and/or tank," which make the headings ambiguous as they were.)

Schedule

The schedule for the program is shown in Figure 1, with shading () through each scheduled activity to show present status.

PROGRESS SUMMARY

Design and Development - Heat Exchanger

The design of the passive heat exchanger module for prototype testing was finalized in March. This includes sizing and selection of the heat exchanger element, air shut-off valve, and filter, as well as cabinet dimensions. The heat exchanger element is a 56-tube parallel series flow arrangement with four tubes in series. The tubes are 1.59 cm O.D. of the tubes. A multi-vane passively controlled air damper was selected. Both of these items are commercially available from suppliers.

Procurement

Orders were placed for the heat exchanger element, air damper, air filter and panel sections of the cabinet. Delivery is expected by late April for all purchased items.

Records and Documentation

Engineering drawings were completed for the passive heat exchanger subsystem and components. Special Manufacturing Instructions and Required Materials and Parts document preparation was begun as well as the Installation, Operation and Maintenance Manual draft. These will be completed in preparation for the Prototype Design Review on April 27, 1977.

Schedule

The schedule for the program is shown in Figure 1. Shading () through each scheduled activity shows the present status of that activity.

Other

A paper entitled "A Passive Thermosyphon Heat Exchanger for Residential Solar Applications" by J. L. Franklin, E. W. Saaski (both of Sigma Research, Inc.) and J. D. Hankins (NASA-MSFC) was prepared and submitted for the Annual Meeting of the International Solar Energy Society at Orlando, Florida on June 6-10, 1977.

8-1

THIRD QUARTERLY REPORT

PROGRESS SUMMARY

Engineering

Analysis of the thermal performance qualification tests has been completed and a report prepared to present these results. That report is transmitted under separate cover. The system performance exceeded contract specifications as illustrated by the data comparison in attached Figure 1. The use of a direct drive blower was shown to result in 20% less electrical power input to the blower with equal or improved air handling capabilities.

Procurement

Purchase orders were issued for all parts needed to fabricate the three deliverable Maxi-therm S-101 assemblies. Longer lead items include the heat exchanger coil and the air damper. These are scheduled for early September delivery. No problems of material delivery to meet the fabrication schedule are indicated.

Documentation

Preparation of the Qualification Test Report was completed and preparation of the Acceptance Test Procedure Plan was also completed. These two documents were transmitted to NASA Marshall under separate cover.

Schedule

The schedule for the program is shown in Figure 2 with shading to show present status of each activity.

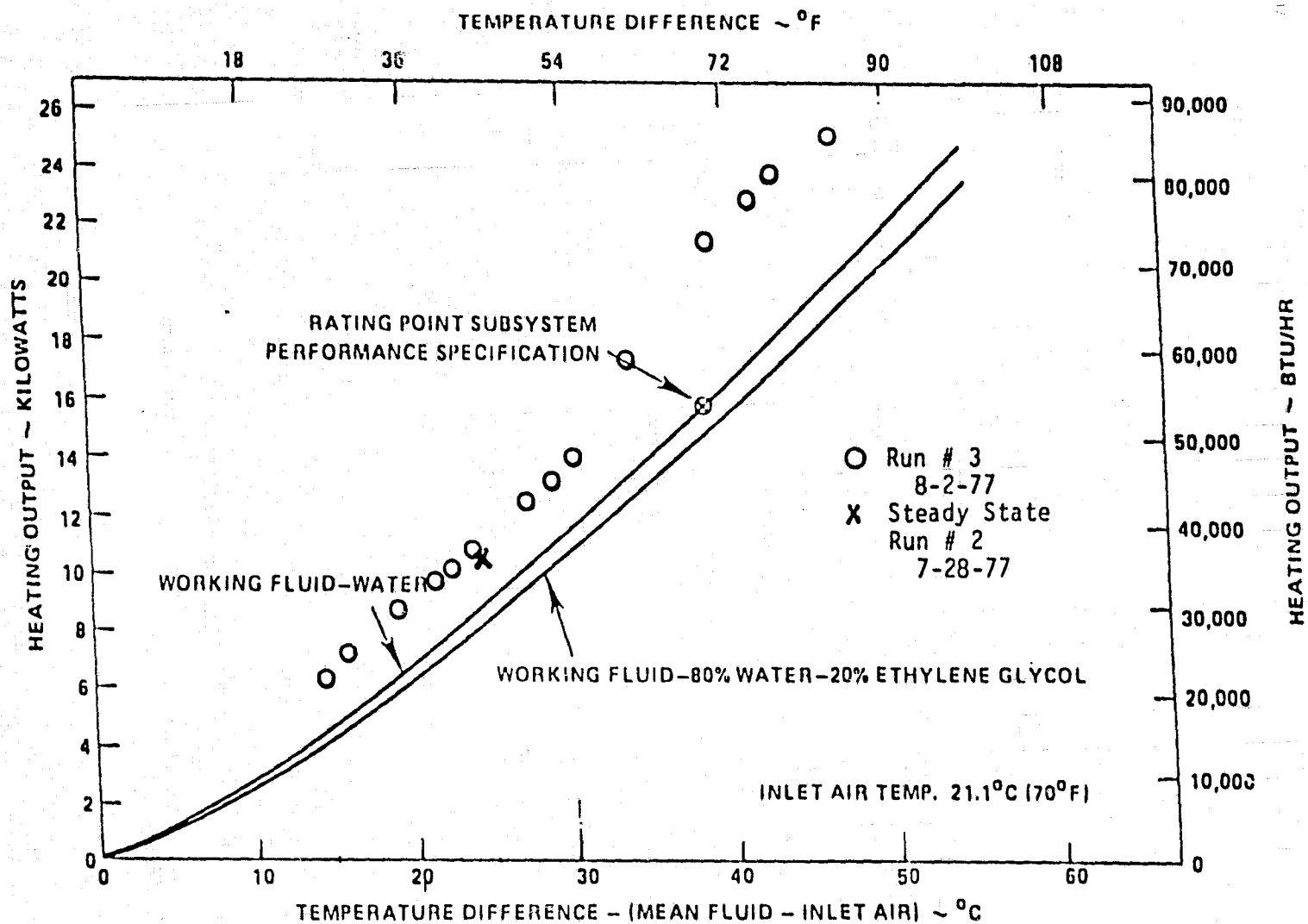
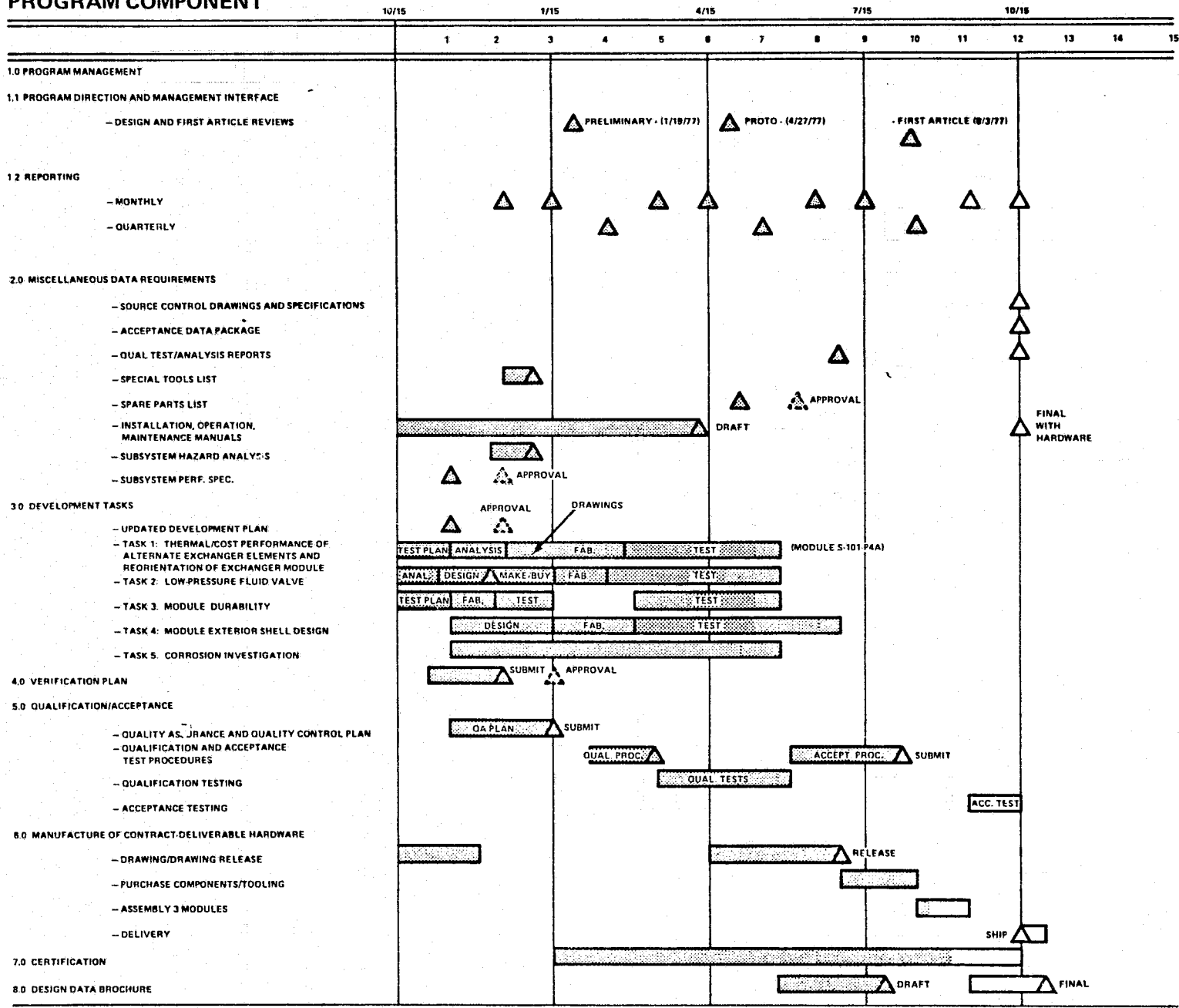


FIGURE 1. MINIMUM PERFORMANCE CHARACTERISTICS OF MAXI-THERM MODEL S-101. THE POINT DENOTED ⊗ IS THE SUBSYSTEM PERFORMANCE SPECIFICATION RATING POINT TO BE VERIFIED BY TEST. THE REMAINING PORTIONS OF LINES ARE TO BE DEMONSTRATED BY ANALYSIS BASED ON THE TEST RATING POINT.

PROGRAM COMPONENT



8-4

FIGURE 2. PROGRAM SCHEDULE FOR SOLAR HEATING MODULE WITH SUPPLEMENTARY HEATING (FOR USE IN SINGLE FAMILY BUILDINGS) (CONTRACT TERM 10/30/76 TO 10/30/77)

APPENDIX A

The following listings are for the computer codes used in the design and evaluation of the parallel and parallel-series circuited heat exchangers.

The first program, MSFC1, is for the former circuiting system and the second, MSFC2, for the latter.


```

MSFC1
10 DEF FNG1(A)=1./((2.1482*((A-8.435)+(8078.4+(A-8.435)**2)**.5)-120.)) !VISC-AI
E
20 DEF FNG2(B)=7.9067E-9*(B+273.15)**2.73135*EXP(-6.7275E-3*(B+273.15)) !T. C.
30 DEF FNG4(C)=8.958663-.04053396*(273.15+C)+1.12431E-4*(273.15+C)**2-1.01379E
*(273.15+C)**3 !CP OF H2O--W-SEC/GM-C
40 R5=1. !NOMINAL DENSITY OF H2O-GM/CC
50 DIM Q(20),W3(20)
60 P2=3.141592654
70 !-----EXCHANGER MODULE SPECIFICATIONS-----
80 READ T4,T5 !AMBIENT AIR AND TANK OUTLET TEMPERATURE-C
90 DATA 21.11,60.
100 READ S,T !FIN INNER SPACING AND THICKNESS-CM
110 DATA .2286,.0254
120 READ D,W !TUBE O.D. AND WALL THICKNESS-CM
130 DATA 1.905,.0508
140 R=D/2.-W
150 READ C4,A1 !FIN LENGTH-CM AND CENTERLINE-TO-CENTERLINE SPACING RATIO
160 DATA .9525,2.
170 READ A2 !FIN THERMAL CONDUCTIVITY-W/CM-C
180 DATA 2.04
190 READ C5,N1 !NOMINAL NUMBER OF TUBES WIDE AND EXACT NUMBER DEEP
200 DATA 12.,3.
210 READ M1 !ACTUAL NUMBER OF TUBES
220 DATA 35
230 READ C2 !FINNED TUBE LENGTH-CM
240 DATA 60.96
250 READ L9 !DRIVING PRESSURE HEIGHT-CM
260 DATA 121.92
270 !-----INLET AND OUTLET PIPING-----
280 READ D2 !PIPE I.D.-CM
290 DATA 10.16
300 READ L1,L2 !INLET AND OUTLET LENGTH-CM
310 DATA 60.96,228.6
320 READ R2 !ELBOW RADIUS-CM
330 DATA 22.86
340 !-----AIR PROPERTIES-----
350 READ C1,P1,C7,P3 !T.C.--W/CM-C--PRANDTL--VISC.--POISE--DENSITY-GM/CC
360 DATA 2.558E-4,.72,1.833E-4,.0012
370 READ C3 !AIR FLOW RATE-GM/SEC
380 DATA 930.7
390 C6=((A1-1.)*D-2.*C4*T/(S+T))*C5*C2
400 C8=C3*D/(C6+C7)
410 C9=C3/C6
420 !BRIGGS AND YOUNG CORRELATION
430 H1=.134*(C1/D)*P1**-.33*C8**-.681*(S/C4)**.2*(S/T)**.1134
440 A3=P2*C2*(S*D+((D+2.*C4)**2-D**2)/2.)/(S+T)
450 H=A3*H1
460 !FIN PARAMETERS TO DETERMINE EFFICIENCY
470 A4=(C4+T/2.)*.5*(2.+H1/(A2+T*C4))*+.5
480 A5=(D/2.+C4+T/2.)/(D/2.)
490 PRINT A4,A5
500 INPUT "THE FIN EFFICIENCY IS";E
510 H2=H+E
520 !BEGIN ITERATIVE PROCEDURES
530 INPUT "THE INITIAL GUESS ON THE OUTLET TEMPERATURE AND FLOW RATE ARE";T1,M1
540 T2=.5*(T1+T5)
550 P8=(T1-T5)*(.0142508+3.11471E-5*(T2+273.15)-1.378E-3*(T2+273.15)**.5) !C
RHO/DEL T)*DEL T--GM/CC

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60 P9=L9+R9+R81. !DRIVING PRESSURE-DYNES/CM2
70 !-----CONSTANTS IN VARIOUS SYSTEM PRESSURE DROPS-----
80 V1=.298/(D2**4)
90 V2=.24113+L1+FN61(T5)**.25/(D2**4.75) !INLET TUBE-FLOW
00 V3=1.2159/(M1**2*(2.+P)**4) !FINNED TUBES-ENTRANCE
10 V4=.24113+D2+FN61(T2)**.25/((2.+R)**4.75+M1**1.75) !FINNED TUBES-FLOW
20 V5=.24113+L2+FN61(T1)**.25/(D2**4.75) !OUTLET TUBES-FLOW
30 V6=.7154+R2**+.9+FN61(T1)**.2/(D2**4.7) !OUTLET ELBOW-TUBE & HEADER TO TUBE
40 FOR N=1 TO 50
50 M2=(P9+R5/(M1**+.25+V1+V2+M1**+.25+V3+V4+V5+M1**+.05+V6))**+.57143
60 IF (M2/M1)<.999 GOTO 690
70 IF (M2/M1)>1.001 GOTO 690
80 GOTO 710
90 M1=M2
00 NEXT N
10 R9=2.*(M2/W1)/(P2+R+FN61(T2)) !REYNOLDS NUMBER IN FINNED TUBES
20 P8=FN61(T2)+FN64(T2)/FN62(T2) !PRANDTL NO.
30 N2=.0155+R9**+.83+P8**+.5 !TURB. EQN. FROM KAYS AND DATA
40 H3=P2+D2+FN62(T2)+N2 !H+A FOR TUBE INTERIOR
50 A8=1./(1./H3+1./H2) !SERIES CONDUCTANCE
60 W2=T4 !DUMMY INLET AIR TEMPERATURE
70 W4=0. : W7=0.
80 FOR M=1 TO N1
90 Q(M)=A8+(T5-W2)/(1.+(A8/2.)*(W1/(M2+FN64(T2))+C5/(C3+1.)))
00 IF W7=1. GOTO 820
10 W6=C5 : W7=1. : GOTO 830
20 W6=C5-1. : W7=0. : GOTO 830
30 W2=W2+Q(M)+W6/(C3+1.) !UPDATE AIR TEMPERATURE
40 W3(M)=T5-Q(M)+W1/(M2+FN64(T2)) !CALCULATE WATER OUTLET TEMPERATURE
50 W4=W4+W6+W3(M)/W1 !CALCULATE AVERAGE WATER OUTLET TEMPERATURE
60 NEXT M
70 PRINT W4
80 IF ABS(T1-W4)<.01 GOTO 910
90 T1=W4 : M1=M2 : GOTO 540
00 !-----SYSTEM PRESSURE DROPS-----
10 U1=V1+M2**2/R5
20 U2=V2+M2**1.75/R5
30 U3=V3+M2**2/R5
40 U4=V4+M2**1.75/R5
50 U5=V5+M2**1.75/R5
60 U6=V6+M2**1.8/R5
70 PRINT : PRINT "WATT OUTPUTS PER TUBE FOR EACH ROW" : MAT PRINT Q(N1),
80 PRINT : PRINT "H+A(TUBE)-H+A(FIN)-MDOT-H+A(TOT)-DELP(DRIVE)-RE-NU-DEL RHO"
90 PRINT H3,H2,M2,A8,P9,R9,N2,R8
000 PRINT : PRINT "SYSTEM PRESSURE DROPS"
010 PRINT U1,U2,U3,U4,U5,U6
020 !BRIGGS AND ROBINSON CORRELATION
030 F=18.93/(C8**+.316+A1**+.927)
040 P4=F+C9**2+N1/P3
050 P7=P4+.00209/5.2 !PRESSURE DROP IN IN.WG
060 P5=(C2/(S+T))*P2+((2.+D4+D)**2-D**2)*T+C5/4.
070 P6=W+P2+D+C2+C5
080 PRINT : PRINT "DEL P AIR(IN. WG)"
090 PRINT P7
100 INPUT "SHOULD I CONTINUE 1=YES 2=NO":K
110 IF K>1 GOTO 1130
1120 GOTO 520
1130 END

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MS/C2
10 DEF FNG1(A)=1./((2.1482*((A-8.435)+(8078.4+(A-8.435)**2)**.5)-120.) !VISC-POI
E
20 DEF FNG2(B)=7.9067E-9*(B+273.15)**2.73135*EXP(-6.7275E-3*(B+273.15)) !T. COE
30 DEF FNG4(C)=8.958663-.04053396*(273.15+C)+1.12431E-4*(273.15+C)**2-1.01379E-
*(273.15+C)**3 !CP OF H2O---W-SEC/GM-C
40 R5=1. !NOMINAL DENSITY OF H2O-6M/CC
50 DIM Q(20),M3(20)
60 P2=3.141592654
70 !-----EXCHANGER MODULE SPECIFICATIONS-----
80 READ T4,T5 !AMBIENT AIR AND TANK OUTLET TEMPERATURE-C
90 DATA 21.11,60.
100 READ S,T !FIN INNER SPACING AND THICKNESS-CM
110 DATA .2337,.02032
120 READ D,W !TUBE O.D. AND WALL THICKNESS-CM
130 DATA 1.27,.0508
140 R=D/2.-W
150 READ C4,A1 !FIN LENGTH-CM AND CENTERLINE-TO-CENTERLINE SPACING RATIO
160 DATA .635,2.
170 READ A2 !FIN THERMAL CONDUCTIVITY-W/CM-C
180 DATA 2.04
190 READ C5,N1 !NOMINAL NUMBER OF TUBES WIDE AND EXACT NUMBER DEEP
200 DATA 24.,3.
210 READ M1 !ACTUAL NUMBER OF TUBES
220 DATA 71.
230 READ C2 !FINNED TUBE LENGTH-CM
240 DATA 60.96
250 READ L9 !DRIVING PRESSURE HEIGHT-CM
260 DATA 121.92
270 READ R3 !RETURN BEND(180 DEG)RADIUS
280 DATA 1.905
290 !-----INLET AND OUTLET PIPING-----
300 READ D2 !PIPE I.D.-CM
310 DATA 7.62
320 READ L1,L2 !INLET AND OUTLET LENGTH-CM
330 DATA 60.96,228.6
340 READ R2 !ELBOW RADIUS-CM
350 DATA 22.86
360 !-----AIR PROPERTIES-----
370 READ C1,P1,C7,P3 !T.C.--W/CM-C--PRANDTL--VISC.--POISE--DENSITY-GM/CC
380 DATA 2.558E-4,.72,1.833E-4,.0012
390 READ C3 !AIR FLOW RATE-GM/SEC
400 DATA 930.7
410 C6=((A1-1.)*D-2.*C4+T/(S+T))*C5+C2
420 C8=C3*D/(C6+C7)
430 C9=C3/C6
440 !-----BRIGGS AND YOUNG CORRELATION-----
450 H1=.134*(C1/D)*P1**.33+C8**.681*(S/C4)**.2*(S/T)**.1134
460 A3=P2+C2*(S+D+((D+2.*C4)**2-D**2)/2.)/(S+T)
470 H=A3+H1
480 !-----FIN PARAMETERS TO DETERMINE EFFICIENCY-----
490 A4=(C4+T/2.)*.5*(2.*H1/(A2+T+C4))***.5
500 A5=(D/2.+C4+T/2.)/(D/2.)
510 PRINT A4,A5
520 INPUT "THE FIN EFFICIENCY IS":E
530 H2=H+E
540 !-----BEGIN ITERATIVE PROCEDURES-----
550 INPUT"THE INITIAL GUESS ON THE OUTLET TEMPERATURE AND FLOW RATE ARE":T1,M1
560 T2=.5*(T1+T5)
570 P8=(T1-T5)*(.0142508+3.11471E-5*(T2+273.15)-1.378E-3*(T2+273.15)**.5) !DEL

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RHO/DEL T) * DEL T -- GM/CC
380 P9=L9+P8*981. !DRIVING PRESSURE-DYNES/CM2
390 !-----CONSTANTS IN VARIOUS SYSTEM PRESSURE DROPS-----
400 V1=.292/(D2**4) !ROUNDED INLET PIPING ENTRANCE
410 V2=.24113*L1*FNG1(T5)**.25/(D2**4.75) !INLET TUBE-FLOW
420 V3=1.2159/(C5**2*(2.*R)**4) !FIN TUBE INLET AND OUTLET
430 V4=.24113*C2*N1*FNG1(T2)**.25/((2.*R)**4.75*C5**1.75) !FIN TUBES-FLOW
440 V7=.7154*(N1-1)*R3**.9*FNG1(T2)**.2/((2.*R)**4.7*C5**1.8) !RETURN BENDS
450 V5=.24113*L2*FNG1(T1)**.25/(D2**4.75) !OUTLET TUBES-FLOW
460 V6=.7154*R2**.9*FNG1(T1)**.2/(D2**4.7) !OUTLET ELBOW-TUBE & HEADER TO TUBE
470 FOR N=1 TO 50
480 M2=(P9+R5/(M1**.25*V1+V2+M1**.25*V3+V4+V5+M1**.05*V6+M1**.05*V7))**.57143
490 IF (M2/M1) < .999 GOTO 720
500 IF (M2/M1) > 1.001 GOTO 720
510 GOTO 740
520 M1=M2
530 NEXT N
540 R9=2.*(M2/C5)/(P2+R*FNG1(T2)) !FIN TUBE REYNOLDS NUMBER (LIQUID)
550 P8=FNG1(T2)*FNG4(T2)/FNG2(T2) !PRANDTL NO.
560 N2=.0155*R9**.83+P8**.5 !TURB. EQN. FROM KAYS AND DATA
570 H3=P2+C2*FNG2(T2)*N2 !H*A FOR TUBE INTERIOR
580 A8=1./(1./H3+1./H2) !SERIES CONDUCTANCE
590 W2=T4+M2*FNG4(T2)*(T5-T1)/(C3+1.) : W4=T5 !DUMMY OUTLET AIR AND INLET H2O
600 FOR M=1 TO N1
610 Q(M)=A8*(W4-W2)/(1.+(A8*C5/2.)*(1./(C3+1.)-1./(M2*FNG4(T2))))
620 W2=W2-Q(M)*C5/(C3+1.)
630 W3(M)=W4-Q(M)*C5/(M2*FNG4(T2)) !UPDATE WATER TEMPERATURE
640 W4=W3(M)
650 NEXT M
660 PRINT W4,W2
670 IF (W4/T1) < .999 GOTO 900
680 IF (W4/T1) > 1.001 GOTO 900
690 GOTO 920
700 T1=.5*(T1+W4) : M1=M2 : GOTO 560
710 !-----SYSTEM PRESSURE DROPS-----
720 U1=V1*M2**2/R5
730 U2=V2*M2**1.75/R5
740 U3=V3*M2**2/R5
750 U4=V4*M2**1.75/R5
760 U7=V7*M2**1.8/R5
770 U5=V5*M2**1.75/R5
780 U6=V6*M2**1.8/R5
790 PRINT : PRINT "WATT OUTPUTS PER TUBE FOR EACH ROW" : MAT PRINT Q(M1),
1000 PRINT : PRINT "H*A(TUBE)-H*A(FIN)-MDOT-H*A(TOT)-DELP(DRIVE)-RE-NU-DEL RHO"
1010 PRINT H3,H2,M2,A8,P9,R9,N2,R8
1020 PRINT : PRINT "SYSTEM PRESSURE DROPS"
1030 PRINT U1,U2,U3,U4,U7,U5,U6
1040 !-----ROBINSON AND BRIGGS CORRELATION-----
1050 F=18.93/(C8**.316*A1**.927)
1060 P4=F*C9**2*N1/P3
1070 P7=P4+.00209/5.2 !PRESSURE DROP IN IN.WG
1080 P5=(C2/(S+T))*P2*((2.*C4+D)**2-D**2)*T*C5/4.
1090 P6=W*P2*D*C2*C5
1100 PRINT : PRINT "DEL P AIR (IN. WG)"
1110 PRINT P7
1120 INPUT "SHOULD I CONTINUE 1-YES 2-NO":K
1130 IF K>1 GOTO 1150
1140 GOTO 540
1150 END

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